

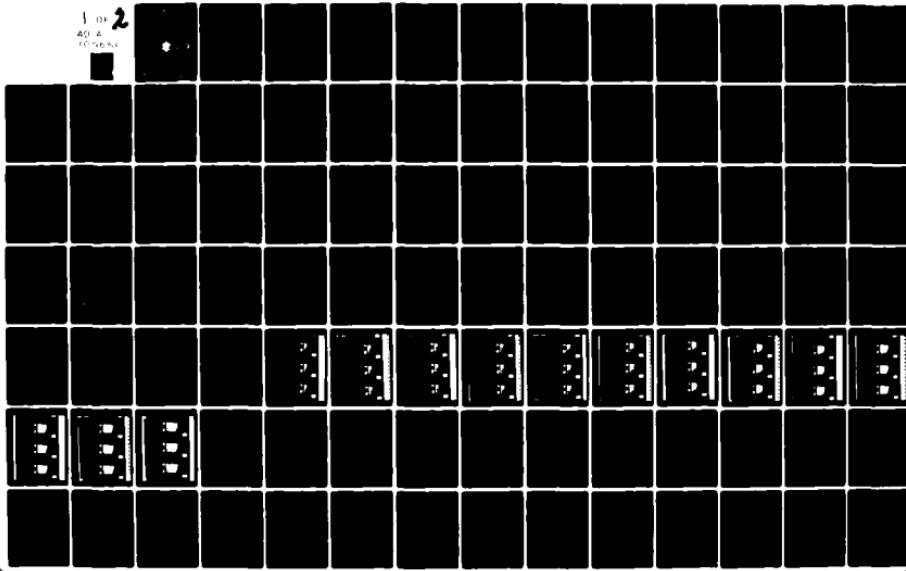
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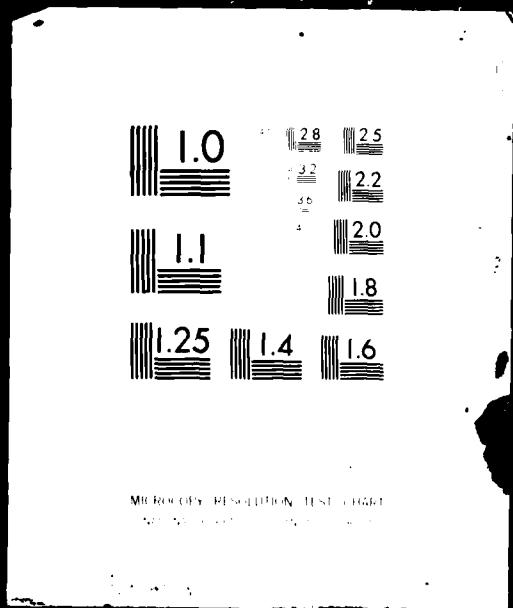
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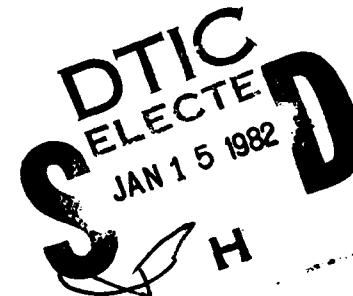
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PROJECT REPORT

NO. 112

"OPTIMIZING THE NAHBE PISTON CAP DESIGN
UTILIZING SCHLIEREN PHOTOGRAPHY METHODS AND
APPLICATIONS OF THE HELMHOLTZ RESONATOR THEORY"



UNITED STATES NAVAL ACADEMY
ANNAPOLIS, MARYLAND
1981

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required. One possible solution to these problems was to model the NAHBE piston cap through the Helmholtz Resonator Theory. This theory generalizes the time dependent reaction of a volume of fluid in a chamber by the acoustic analog to the mechanical system forced oscillator. The theory enables one to predict or design the acoustic impedance of a cavity and its natural frequency by way of the cavity and inlet geometry. This study shows how the solution to the Helmholtz Theory may be obtained and the fluid mass displacement of NAHBE cap modeled by the Helmholtz Theory predicted. The solution of the NAHBE cap reaction by the Hemholtz Theory, however, is only a hypothesis until it is substantiated. Conclusive evidence is shown in this study through a high-speed Schlieren film analysis of the actual combustion process in a NAHBE, that the theory holds true for and can predict reality. Thus, it is proven that the Helmholtz Theory, which can be used as a design tool, can model the real systems found in heat balanced engines today.

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"Optimizing the NAHBE Piston Cap Design
Utilizing Schlieren Photography Methods and
Applications of the Helmholtz Resonator Theory"

A Trident Scholar Project Report

by

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ABSTRACT

The Naval Academy Heat Balanced Engine (NAHBE) has been studied extensively for the last five years. Efforts to understand and predict the process occurring in the engine were complicated and difficult to verify experimentally. Optimization of the output of the engine, though possible, took much too long a time through parametric variation. An answer to the problem of predicting the time dependent reaction occurring in the engine was sought. Also, some easy design tool for optimizing the reaction was required. One possible solution to these problems was to model the NAHBE piston cap through the Helmholtz Resonator Theory. This theory generalizes the time dependent reaction of a volume of fluid in a chamber by the acoustic analog to the mechanical system forced oscillator. The theory enables one to predict or design the acoustic impedance of a cavity and its natural frequency by way of the cavity and inlet geometry. This study shows how the solution to the Helmholtz Theory may be obtained and the fluid mass displacement of NAHBE cap modeled by the Helmholtz Theory predicted. The solution of the NAHBE cap reaction by the Helmholtz Theory, however, is only a hypothesis until it is substantiated. Conclusive evidence is shown in this study through a high-speed Schlieren film analysis of the actual combustion process in a NAHBE, that the theory holds true for and can predict reality. Thus, it is proven that the Helmholtz Theory, which can be used as a design tool, can model the real systems found in heat balanced engines today.

NOMENCLATURE

| | |
|----------------|--|
| A | Combustion cylinder cross-sectional area |
| a | Opening radius, Speed of Sound |
| B | Damping constant |
| C | Acoustic capacitance, capacitance |
| c | Speed of sound |
| g | Gravitational constant |
| H | Combustion cylinder length |
| i | Current |
| K ₁ | Runge-Kutta constant |
| k | Ratio of specific heats |
| l | Cap thickness, neck length |
| l' | Effective length |
| l _i | Runge-Kutta constant |
| M | Mass volume, mass element |
| P | Pressure |
| Q | Quality number |
| R | Acoustic resistance, resistance |
| S | Opening cross-sectional area |
| T | Temperature |
| t | Time |
| u | Volume velocity |
| v | Volume, voltage |
| x | Volume displacement, displacement |
| x | Variable |
| y | Variable |
| z | Acoustic Impedance |

z displacement
 γ Compression ratio
 ζ Non-dimensional distance
 ρ_0 Density
 τ Non-dimensional time
 ω Frequency

INTRODUCTION

Growing concern over rising petroleum fuel prices has increased the interest in economical consumption. To solve the long term problem of limited resources it will be necessary to do away with common internal combustion engines and all other power sources requiring fossil fuels. In the interim, however, it will be necessary to stretch out what reserves we do have so that technology and science will have time to find a solution to the problem.

Little had been done before the 1970's to increase fuel economy except in the field of aviation where extra fuel meant less payload. The basic design and concept was still never improved, however. Two thermodynamic models for internal combustion cycles that have been known for some time are the air standard Otto and Diesel cycles. The Otto cycle is a process which uses a heat addition at constant volume to result in work. The Diesel cycle, on the other hand, is a constant pressure heat addition process. A combination of these two cycles can be done in equilibrium or quasi-equilibrium. The former is known as the classic Dual cycle. The quasi-equilibrium cycle was found to be desirable because of theoretical increases in fuel efficiency and decreases in peak pressures (Reference 1). The quasi-equilibrium cycle, however, must meet certain limitations, making it seem impractical at first glance.

The thermodynamic model for the quasi-equilibrium process, known as the Heat Balanced cycle, is summarized in figure 1 and table I (Reference 2). Two points should be noted. Firstly, the total mass is separated into two regions. One region has heat added in a constant volume phase (Otto cycle) and the other has heat addition during a constant pressure phase (Diesel cycle). Secondly, the two regions must be kept separate and nonmixing if quasi-equilibrium is to exist.

The Naval Academy Heat Balanced Engine (NAHBE) obtains the desired results by placing a cap on a normal piston so as to create a chamber of gas partially separated from the combustion chamber as illustrated in figure 2. The results of a parametric investigation showed that a performance giving more than a 30 percent increase in power was achievable. Furthermore, fuel consumption could be improved by 10 percent and the CO emissions could be cut to .1 percent and unburnt hydrocarbons to 100 ppm, excellent by any standards (Reference 3). The precise reasons why this type of configuration has such results and what the optimizing parameters are not understood. Past optimization of the effect of the balancing chamber for certain engines has been found through "a long and tedious process" of parametric variation (Reference 3). Obviously, this type of guess work must be eliminated if the concept of heat balancing is to be feasible for utilization in basic designing of engines.

An early step in understanding the process associated with the Heat Balanced cycle was a non-steady flow analysis of the NAHBE combustion process completed in 1978 by Raghavan P. Pandalai. Using classical one-dimensional non-steady flow solution techniques, Pandalai found the exact time solution of the flow of a mass into and out of an idealized balancing chamber. This solution is shown in figure 3. The interface at non-dimensional time zero is simply the assumed separation point of combustible gases and air. Once combustion is begun, a shock wave is transmitted into the balancing chamber which accelerates the interface to motion towards the chamber. The shock wave rebounds from different cross-sectional area reductions at non-dimensional distances equal to 1.0, 1.78 and 2.2. The most important point is that the interface is slowed and finally reversed in direction (region 24 and 31). This indicates there might be some type of oscillatory motion of the mass volume into and out of the balancing chamber (Reference 4). It has been

suggested that this oscillatory motion could be predicted and the exact non-steady solution for the mass flow generalized by the Helmholtz Resonator Theorem.

THE HELMHOLTZ RESONATOR THEOREM

A Helmholtz Resonator may be visualized as a bottle with a neck as illustrated in figure 4. The volume of fluid in the chamber is disturbed by pressure waves from some source (imagine a small speaker humming at one particular frequency). The fluid medium is alternately compressed and expanded by the pressure waves as they move into the opening. The fluid in the opening may be considered to act as a mass element, a piston. There is a stiffness related to the size of the opening and a resistance to motion associated with the radiation of energy in the form of sound. Other resistive forces, such as viscosity, will not be treated in this discussion. This system may be modeled by an analogous mechanical system of the type shown in figure 5. The equation describing the displacement of the mass of such a forced mechanical oscillator is

$$(M)\ddot{z} + (B)\dot{z} + (K)z = f \quad (1)$$

Now the mass, M, stiffness, K, and damping, B, must be described in terms of the acoustic system.

The mass of the mechanical system is transposed in acoustic terms to $\rho_0 S l'$, where ρ_0 is the density of the gas, S, the cross-sectional area of the opening, and l' the effective length of the opening. This effective length is greater than the physical length of the resonator, l, and related by the diameter of the opening, 2a, through the equation

$$l' = l + \Delta l \quad (2)$$

where $\Delta l = .85a$ for wide flanged constrictions and $\Delta l = .6a$ for unflanged openings. The spring constant, K in acoustic terms is $\rho_0 c^2 S^2 / V$ where c is the speed of sound through the gas and V is the volume of the chamber. The damping constant or effective resistance, B, is found to be $\rho_0 c k^2 S^2 / 2\pi$ in terms of acoustic variables k, the ratio of specific heats for the gas, and

the others already defined. Finally, the driving force, f , is produced by the instantaneous acoustic pressure wave of amplitude P as it impinges on the cross-sectional area S . This term is $SP \cos(\omega t)$. Substituting terms, the differential equation describing the inward displacement of the fluid, y , is

$$\rho_0 \left[\frac{Sd^2y}{dt^2} + \frac{\rho_0 ck^2 S^2}{2\pi} \frac{dy}{dt} + \frac{\rho_0 c^2 S^2}{V} y \right] = SP \cos(\omega t) \quad (3)$$

Just as the acoustic system was modeled by a mechanical system, the mechanical system is easily modeled by an electrical system such as the one in figure 6. In this case, the mass, M , or the volume velocity, u , for an acoustic system which is found through the volume displacement, X , by the relationship $u = \frac{dx}{dt}$, is analogous to the current of the electrical system. The voltage corresponds to the mechanical and acoustic systems by $V = P \cos(\omega t)$. The equation describing the electrical system made up on an inductive element, L , a resistive element, and a capacitive element of magnitudes M , R , C respectively is

$$\frac{Mdi}{dt} + Ri + \frac{1}{C} idt = V \quad (4)$$

Again, the problem reduces to solving the electrical constants in terms of the acoustic constants.

In order to do this, inspection of the equation describing the mechanical system is in order. Using the equations

$$X = yS \quad (5)$$

$$\frac{dX}{dt} = \frac{dy}{dt} S \quad (6)$$

equation 3 reduces to

$$\rho_0 \left[\frac{d^2X}{dt^2} + \frac{\rho_0 ck^2 S}{2\pi} \frac{dX}{dt} + \frac{\rho_0 c^2 S}{V} X \right] = SP \cos(\omega t) \quad (7)$$

and further to

$$\frac{\rho_0 l'}{S} \frac{d^2X}{dt^2} + \frac{\rho_0 c k^2}{2\pi} \frac{dX}{dt} + \frac{\rho_0 c^2}{V} X = P \cos(\omega t) \quad (8)$$

Finally, since

$$i = u = \frac{dX}{dt} \quad (9)$$

and

$$V = P \cos(\omega t) \quad (10)$$

then relating terms in equations 4 and 8 it is obvious that

$$M = \rho_0 l'/S \quad (11)$$

$$R = \rho_0 c k^2 / 2\pi \quad (12)$$

$$\frac{1}{C} = \rho_0 c^2 / V \quad (13)$$

Thus the acoustic impedance of a Helmholtz Resonator may be found through the electrical analogy to be

$$Z = R + j(\omega M - \frac{1}{\omega C}) \quad (14)$$

Equation 9 points to the fact that there is some frequency, ω_0 , that results in the imaginary term of the acoustic impedance to be zero and resonance to occur, hence the name Helmholtz resonator. This frequency is found easily by,

$$\omega_0 M - \frac{1}{\omega_0 C} = 0 \quad (15)$$

then

$$= (1/MC)^{1/2} = c(S/l'V)^{1/2} \quad (16)$$

The similarity to the electrical system may be carried even further to the actual frequency response of the system. That is the quality and sharpness of resonance may be predicted by the analogy to the electrical Q point. That is

$$Q = \frac{\omega_0 M}{R} = 2\pi(l'^3 V / S^3)^{1/2} \quad (17)$$

It is vital to note three important points about the preceding discussion (Reference 5).

Firstly, the only easily adjustable variables of the resonator are those associated with its geometry. That is the gap diameter, the volume, cross-sectional area and the length of the entrance to the chamber. The theory is completely general as to the shape of the volume except as to a flanged or unflanged entrance. Secondly, both the resonant frequency and quality of response depend only on geometric variables. Finally, this theory generalizes the exact solution and gives an average of the fluid displacement and velocity. Thus, such tedious methods of solution as noted earlier may be side stepped.

The applications of the Helmholtz Theory to the NAHBE should be apparent upon comparison of the balancing chamber in figure 2 and Helmholtz resonator shown in figure 4. The shock wave produced by combustion is a pressure differential wave and can be considered the driving wave as it rebound back and forth from the balancing chamber to the top of the combustion chamber and back into the balancing chamber (Reference 6).

PROBLEM STATEMENT AND METHOD OF SOLUTION

If the Helmholtz Resonator Theory is to be accepted as a model for the NAHBE, it is necessary to substantiate it. In order to accomplish the goal of correlating the Helmholtz Resonator Theory and the actual combustion cycle of the NAHBE, it is primarily necessary to find some way to solve the Helmholtz equation for the volume displacement and velocity. Once this step is accomplished, the response of the Helmholtz system with parameters of various balancing chamber geometries can be found. This response can then be scrutinized to see if the Helmholtz Theory is a useful and realistic model. The resonant frequency can be readily compared to frequencies of shock waves moving in the combustion chamber.

Secondly, physical data, either quantitative or qualitative, of the actual combustion process can prove invaluable. Such data could substantiate or destroy the correlation of the Helmholtz Theory with reality. Especially useful would be photographs of the combustion process. Such photographs might capture the oscillations and the physical data could be reduced to give both mass displacement and velocities of gases and combustion interfaces.

COMPUTER ANALYSIS

The solution of the Helmholtz Equation, if it was to be practical, must be of such a form as to be useful to some form of computer. A usual method of solving differential equations is the LaPlace transformation technique. Unfortunately, the LaPlace transformation of the Helmholtz Equation was not convenient for evaluation on digital computers (Reference 6). Another solution to second order differential equations that is readily adoptable to digital computers is the Runge-Kutta integration method.

This method was used in the computer analysis of the Helmholtz Resonators of various design parameters. The digital computer analysis was accomplished by the program "SOLVE" and its subprograms "EQMOT" and "VALUES". These programs, written by Ens. J. Brastauskas and Midshipman W. Johnson, may be found in Appendices A, B, and C.

The Runge-Kutta method may be used to determine the solution of any order ordinary differential equation of the form

$$\frac{d^n y}{dx^n} = f(x, y, \frac{dy}{dx}, \dots, \frac{d^{n-1}y}{dx^{n-1}})$$

which is comparable to the equation that results from the Helmholtz Theory. The solution begins by transforming the variable so that only first order equations are obtained. This is done by letting

$$\begin{aligned} Y_1 &= y \\ Y_2 &= \frac{dy}{dx} \\ Y_3 &= \frac{d^2y}{dx^2} \\ Y_n &= \frac{d^{n-1}y}{dx^{n-1}} \end{aligned}$$

Further, define

$$F_1 = \frac{dy_1}{dx} = \frac{dy}{dx} = Y_2$$

$$F_2 = \frac{dy_2}{dx} = \frac{d^2y}{dx^2} = Y_3$$

$$F_n = \frac{dy_n}{dx} = \frac{d^n y}{dx^n} = f(x, Y_1, Y_2, \dots, Y_n)$$

The Runge-Kutta method allows a numerical solution of the differential equation in the form $y(x)$ by approximating this solution over an interval of x by beginning at a specified known starting value for $y(x)$ and using the Runge-Kutta recurrence formula. Thus, given a second order differential equation, such as the Helmholtz Equation,

$$\frac{d^2y}{dx^2} = f(x, y, \frac{dy}{dx})$$

$$Y_1 = y$$

$$Y_2 = \frac{dy}{dx}$$

$$F_1 = Y_2$$

$$F_2 = F(x, Y_1, Y_2) = \frac{d^2y}{dx^2}$$

which gives two ordinary coupled differential equations $F_1 = \frac{dy_1}{dx}$ and $F_2 = \frac{dy_2}{dx}$.

For the Helmholtz Equation, $F_1 = \frac{dx}{dt}$

$$F_2 = d^2x = (-r/m) Y_2 - (1/cm) Y_1 + (p/m) \cos \omega t.$$

The subprogram "EQMOT" gives values for F_1 and F_2 for the given initial conditions and constants.

The solution at particular points after the initial conditions, $x + \Delta x$, are found by the Runge-Kutta equations for a fourth-order differential.

They are:

$$Y_{1,i+1} = \frac{Y_{1i} + K_1 + 2K_2 + 2K_3 + K_4}{6}$$

$$K_1 = f(x_i, Y_{1i}, Y_{2i}) \times$$

$$K_2 = f\left(x_i + \frac{\Delta x}{2}, Y_{1i} + \frac{K_1}{2}, Y_{2i} + \frac{l_1}{2}\right) \Delta x$$

$$K_3 = f\left(x_i + \frac{\Delta x}{2}, Y_{1i} + \frac{K_2}{2}, Y_{2i} + \frac{l_2}{2}\right) \Delta x$$

$$K_4 = f\left(x_i + \Delta x, Y_{1i} + K_3, Y_{2i} + l_3\right) \Delta x$$

$$Y_{2,i+1} = \frac{Y_{2i} + l_1 + 2l_2 + 2l_3 + l_4}{6}$$

$$l_1 = g(x_i, Y_{1i}, Y_{2i}) \Delta x$$

$$l_2 = g\left(x_i + \frac{\Delta x}{2}, Y_{1i} + \frac{K_1}{2}, Y_{2i} + \frac{l_1}{2}\right) \Delta x$$

$$l_3 = g\left(x_i + \frac{\Delta x}{2}, Y_{1i} + \frac{K_2}{2} + \frac{l_2}{2}\right) \times$$

$$l_4 = g\left(x_i + \Delta x, Y_{1i} + K_3, \frac{Y_{2i}}{2} + l_3\right) \Delta x$$

where $f(x, Y_1, Y_2) = \frac{dy}{dx} = F_1$ and $g(x, Y_1, Y_2) = \frac{d^2y}{dx^2} = F_2$

Thus, once an increment, H, is computed in line 420 of "SOLVE", the process is simply a loop, finding each K constant and l constant (lines 500-620) and then finding the values of y and $\frac{dy}{dt}$ at the new point $t + H$ by the formulas given above (lines 630-640, Reference 7).

The subprogram "VALUES" allows the user to input geometric parameters of a given engine. The values for R, M and 1/C are found using thermodynamic principles and plane geometry formulas.

A plot of the final solution for either displacement or velocity versus time is displayed by the library subprogram "TEKSUB3". From these plots the maximum of the third oscillation is taken and the value of the frequency and maximum displacement was put into data file "DATA3" by way of the program "REVAMP" written by Midshipman Johnson (APPENDIX D). The data file is then plotted by the library program TEKGRAF3 and curve fitted to give easily readable data.

PHYSICAL ANALYSIS

In order to obtain data of the combustion cycle, a time dependent film analysis utilizing Schlieren photography techniques was undertaken. In this analysis, a Schlieren apparatus utilizing the Toepler method was used to photograph density gradients occurring during combustion in a heat balanced engine. Such methods of high speed photography in conjunction with Schlieren techniques have only been introduced very recently (Reference 8). A short review of Schlieren methods is in order.

A well known fact of physics is that a ray of light will be deflected from its normal path as it passes through a transparent medium of different index of refraction than that of the medium in which it was originally traveling. The index of refraction is proportional to the density of a medium, and the magnitude and direction of deflection is dependent upon the relative magnitudes of the densities of the medium entered and exited by the ray of light. These principles are the basis of the Schlieren method. A usual Schlieren apparatus is illustrated in figure 7.

A point light source at S, is stationed at a distance equal to the focal distance, f , of a concave mirror, M_1 . This results in a line of parallel ray reflected off the mirror and through the working section. Any disturbances to the density of the medium in the section will cause a deflection of the rays (dotted lines). Any rays that are undisturbed are passed through in a parallel and straight course. Both sets of rays are reflected by another concave mirror, M_2 . The two sets of rays fall on the screen Q. The Toepler method utilizes a knife edge at the focal point of the second mirror. The knife edge effectively blocks out any rays that are bent too far thus creating a shadow. Any light rays bent in the opposite direction cause an addition of light on the screen resulting in a light spot. Thus, any density gradients of the medium may be viewed (Reference 9).

Such an apparatus was constructed for photographing the combustion of the NAHBE. A square engine with two glass walls (fused silicon) enabled the light rays to pass through the section and not be effected by curved glass walls. The engine was run at 1,000 rpm in spark ignition mode and was the test section. The light source was a 500 watt xenon arc lamp focused through a converging lens. A small slit, acting as a horizontal polarizer, was placed just behind the focal point of this lens so as to eliminate any chromatic aberration. The actual set up is illustrated in figure 8 and the test section (square engine) is shown in figure 9.

In order to capture the combustion sequence and slow it down so that useful data might be observed, a high speed, Fastek motion picture camera was used. Previous studies showed that a camera speed of 2,000 frames per second, and a corresponding shutter speed of 1/6,000th of a second were best. This allows a picture to be taken about every .5 milliseconds. A light meter reading of about 15.5 is optimal also (Reference 8).

RESULTS

Computer Analysis

The generalized time response of several Helmholtz resonators having geometries corresponding to several NAHBE caps now in use follows. Figures 10-21 show the time response for frequencies ranging from 3000 to 40,000 radians per second of the square engine with a NAHBE cap having dimensions stated in table II.

Table II

NAHBE Square Engine Parameters

| | |
|----------------------|----------------------|
| Compression ratio | $\gamma = 3.5$ |
| cap thickness | $l = .1$ inch |
| gap | $2a = .0625$ inch |
| cross-sectional area | $S = .101$ sq. in. |
| volume | $V = .393$ cu.in. |
| resonant frequency | $\omega_0 = 23919.2$ |
| quality number | $Q = 6.99$ |

The pressure of the wave was the only problem in the analysis. The value for the pressure of an acoustic wave can be stated in various ways. Therefore, a unit magnitude value of 1 acoustic pressure was used for the pressure value necessary for the solution. This only changes the magnitude of the driver function which, in return, effects the magnitude of the displacement. This is of no consequence, since it is the shape of the time response solution desired and not the magnitude. In any case, these magnitudes may be considered as relative.

After performing similar sets of solutions for various caps, the maximum displacement occurring during the third cycle was plotted against the frequency.

These plots are found in figures 22 through 25. It should be noted that the plot fits were used only to give a general idea of the shape of the curve. A close inspection of these plots show interesting results. The most obvious difference in the plots is the frequency at which the peak displacement occurs. Further scrutinization shows that the peak becomes sharper and slopes more steeply for each cap. Lastly, it is evident that the displacement before the peak is significantly greater than those after. This trend is especially true for the smaller, squatter curves.

Table III shows the computed values of resonant frequency, quality number for balancing chambers whose cap thicknesses were varied. The volume and cross-sectional area remained constant for all the runs.

Table III

| <u>l</u> | <u>Q</u> | <u>ω_0</u> |
|----------|----------|------------------------------|
| .1" | 6.99 | 23919.2 rad/sec |
| .2" | 14.2 | 18199.7 rad/sec |
| .3" | 24.1 | 15267.2 rad/sec |

It is evident, in consideration of the data in table III, that the peaks of the frequency plots occur at resonance for the chambers. This explains the shift in the curves as being due to a change in the resonant frequency due to the variation of the cap thickness. Further investigation shows that the quality number, Q, gives an idea as to the intensity of the response at resonance. A relatively low Q for the $l = .1"$ geometry shows that the frequency width of the response at .707 times the maximum (around .0000277) is approximately from 18000 rad/sec to 27000 rad/sec or a loosely termed bandwidth of 9000 rad/sec. In contrast, the cap of thickness .3" and $Q = 24$ has a bandwidth of 5000 rad/sec.

The Cooperative Fuel Research (CFR) engine has been used in many parametric variation experiments. The cap used in the CFR has a geometry of $l = .0749"$, $S = .7967$ sq. in. and $V = 2.398$ cu. in. resulting in $\omega_0 = 33937.8$ and $Q = .5895$. The same relationships discussed above are concurred by the CFR data. The bandwidth is found to be 14500 rad/sec approximately.

Physical Analysis

The photographic results of the heat balanced cycle in the square engine proved hard to reduce to quantitative data. The subjective points of the film are easily seen when viewed in motion. Unfortunately, limited space allows only one combustion sequence of one of the films taken to be discussed. Eight other films of excellent quality were taken. Such parameters as volume and flanging were varied. The results discussed are typical of all the film analyses done on the square engine.

It should be noted that the photographs appearing in figures 26 through 64 are the negatives of color prints. Therefore, the light portions are dark and the dark portions light. Furthermore, it must be kept in mind that these pictures are blown-up from 16mm film. As a result, the quality of the picture has been reduced. Fortunately, enough detail exists for a reasonable discussion.

The cap used in these films was square in cross-section. The thickness of one flange was approximately .1 in. and the other side was .2 in. The volume and cross-sectional area of the gap were kept constant at .393 cu. in. and .101 sq. in. respectively. This configuration allowed testing and filming of two different geometries under identical conditions. Furthermore, the geometries are the same as those of the computer analysis models.

The sequence begins as spark ignition takes place in the chamber. The flame front can be seen to expand through figure 36. In figure 35 and 36, shadows resembling ripples in a pond can be seen under the left side of the cap. These rings are most likely the reflected shock wave. Unfortunately,

the speed of the shock wave is so fast that it is impossible to tell whether this is the first, second or third reflection. The initial combustion continues to its completion until about figure 50 where it looks as though the combustion is completed. In figure 51, the rings are again seen, but now above the cap on the left side. This reaction, which most likely is the oscillating of the mass of air in the gap, increases until figure 55. The combustion shows regeneration and spreads across the cap in figures 53 through 58. At approximately figure 58 the secondary combustion of the left side meets an impass. This is a result of a very strong reaction occurring from the right chamber at this time. In figure 62 the secondary combustion is all but completed and is finally quenched as the exhaust valve begins to open in figures 63 through 64.

From a subjective point of view, the film does not necessarily substantiate the use of the Helmholtz model. It does show two distinct combustion processes. Also, it shows there is some sort of flow from under the cap causing the secondary reaction. A closer evaluation, however, shows that the film analysis can be used for quantitative data of some kind.

It is known that the combustion cylinder is charged with a propane-air gas solution at approximately bottom dead center. The volume of the cylinder at this point is found by the product of the cross-sectional area and the length of the cylinder (from top of the cap to top of the cylinder). It is also known that the compression ratio (the ratio of volume at bottom dead center and top dead center) is 3.5. Thus, the volume of gas when the piston is at top dead center is the product of the cross-sectional area and the length of the combustion chamber, H. Mathematically stated, this is

$$V_{BDC} = H_{BDC} \times A \quad (18)$$

$$V_{TDC} = H_{TDC} \times A \quad (19)$$

The compression ratio then is

$$\frac{V_{BDC}}{V_{TDC}} = 3.5 = \frac{H_{BDC}}{H_{TDC}} \quad (20)$$

The volume ratio is also associated with the temperature of the gases which undergo isentropic compression or expansion by the thermodynamic formula (Reference 10).

$$T_2 = T_1 (V_1/V_2)^{k-1} \quad (21)$$

All temperatures are in absolute degrees and k is again the ratio of specific heats of the gas undergoing compression or expansion. For the propane-air mixture, it is considered equal to the value for air of 1.4 (Reference 11).

Therefore, given an initial temperature at the beginning of compression, the temperature of the gas may be found at any other time if the volume or length ratio is known. If this procedure is followed for the square engine, and assuming an initial temperature of 76° or $536^\circ R$, the temperature at the end of compression should be about $884^\circ R$. Actually, losses due to non-reversible processes will cause it to be a little higher. It is at this point that combustion begins.

Previous studies have shown a wave of Mach 1.26 is generated by the combustion gases (Reference 4). To be conservative, let us assume the average value of all waves to be a Mach of 1.1. The speed of sound in any gas is associated with the temperature of the gas by the formula (Reference 12).

$$a = (kRgT)^{1/2} \quad (22)$$

where R is the gas constant for any gas and g the acceleration constant of gravity. For Old English units and air the equation reduces to

$$a = 49.01 (T)^{1/2} \quad (23)$$

Knowing the temperature of the gas then, the speed of the wave in feet per

second may be determined. It is then simple arithmetic to find the time required for the wave to begin from the top of the cylinder, reflect from the cap and return to the cylinder top, knowing the distance to the cap. That is the time, t , required to complete this one oscillation is

$$t = \frac{1.1 (12 \text{ in}/\text{ft})(\text{a ft/sec})}{H \text{ in.}} \quad (24)$$

The frequency, in radians per second, of this oscillation is found by

$$\omega = \frac{2\pi \text{ rad/cycle}}{t \text{ sec/cycle}} \quad (25)$$

Thus, the position of the piston can be used to find the frequency of the oscillating wave occurring in the cylinder. This calculation was done, for the square engine, by the computer program "FREQ" (APPENDIX E). The resulting data was placed into a data file "FREQDATA" and plotted (figures 65 and 66).

When these frequencies are compared to those found by the computer analysis, (figures 22 through 24) they are low and should not cause any significant reaction except possibly at the beginning of the combustion. Closer consideration of the problem, however, shows a shortcoming in the previous analysis. The temperature at the end of compression was considered to stay constant throughout the rest of the cycle, but the volume of the cylinder is continually increasing, reducing the temperature by the same formula as before (equation 21). This would result in slower velocities and even lower frequencies. This is evident by the results of the revised program "FREQ" (APPENDIX F) in figures 67 and 68.

One last failacy in the model is apparent. As propane burns in the air, the temperature of the gas in the cylinder will increase. The temperature of combustion for propane is between 1374°R and 1518°R . An average value of 1446°R is assumed. It is obvious that the metal of the cylinder and the

cooling system of the engine will reduce this temperature significantly. To be conservative, again, it is assumed that the temperature is increased by 20 percent of the temperature of combustion, that is the temperature in the cylinder is raised by 290°R throughout the process.

With this last revision of "FREQ" (APPENDIX G), the output is shown in figure 69. When the values of resonant frequencies for the two geometries studied in the film analysis and the chamber lengths for these frequencies are compared, interesting results are found. Such a comparison is compiled in table IV.

Table IV

| <u>L</u> | <u>ω_0</u> | <u>H</u> |
|----------|------------------------------|----------|
| .1" | 23919.2 rad/sec | 1.5" |
| .2" | 18199.7 rad/sec | 1.9" |

Using bottom dead center as a guiding distance, the photographs may be scaled no matter how large or small they are printed. Thus, a chamber length of 1.5 inches is found in figure 54 of the film sequence. The chamber length of 1.9 inches is found at around figure 58. These finds of the computer model in "FREQ" and the subjective data of the film analysis are corroborative. Similar studies on past film analysis obtain comparable results (Reference 8).

SUMMARY OF RESULTS

Computer analysis shows that the square engine caps modeled by the Helmholtz Theory had a correlation between geometry and resonant frequency, quality of reaction, and bandwidth or reaction.

Physical analysis by high speed filming shows a definite correlation between the computer analysis prediction and actual reaction. The film captured every aspect of the combustion process and parts of the wave interaction.

CONCLUSION

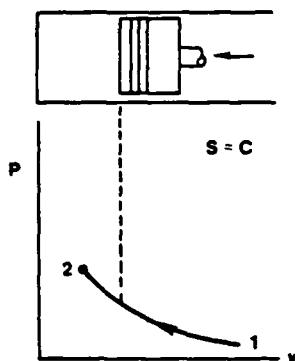
It is evident, by the correlation of the computer and physical data, that the Helmholtz Resonator Theory describes the general time dependent reaction of the NAHBE cap. It may be possible to predict the actual physical reaction occurring during combustion using the Helmholtz Theory. Furthermore, the Helmholtz Theory may be used to optimize the reaction of heat balanced processes now in use in the NAHBE. It may also possibly be used as a design tool to create optimum output given operating specifications of an engine.

Further studies should be made to find the relationship between the intensity of the reaction and output power. That is to say, the study should find which is more desirable, a longer weaker reaction corresponding to a low Q value, or a strong, short reaction typical of high Q values.

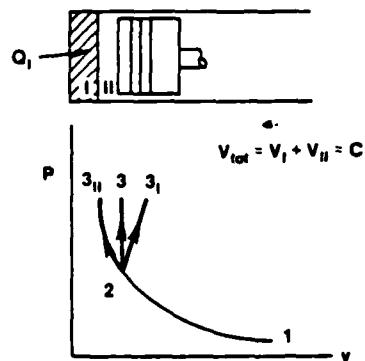
The film analysis of combustion gives valuable insight into the actual process. Quantitative, as well as qualitative, data may be obtained and used in future studies. Reasons for such side effects as low hydrocarbon emissions are evident by viewing the extremely long combustion of the gases. Film analysis has been refined to the point that it can easily be used for any type of future research.

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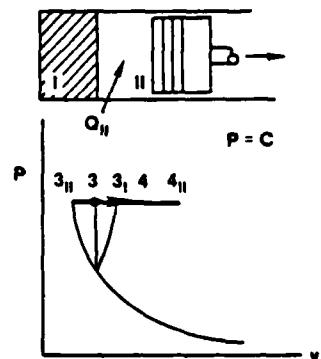
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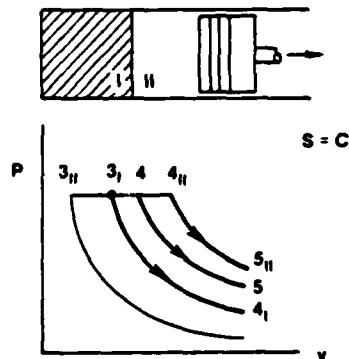
a) compression process



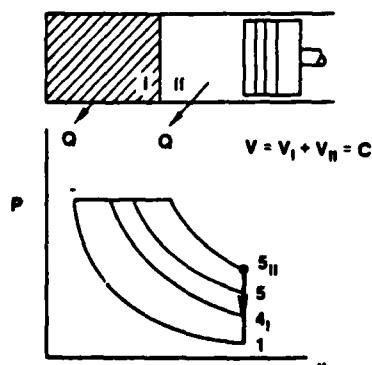
b) first heat addition process



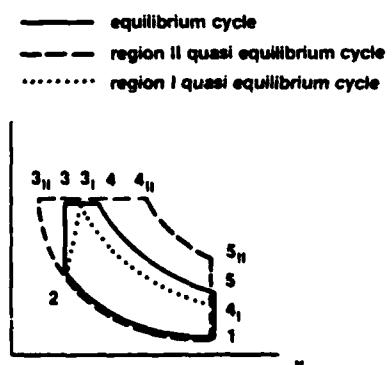
c) second heat addition process



d) expansion process



e) heat rejection process



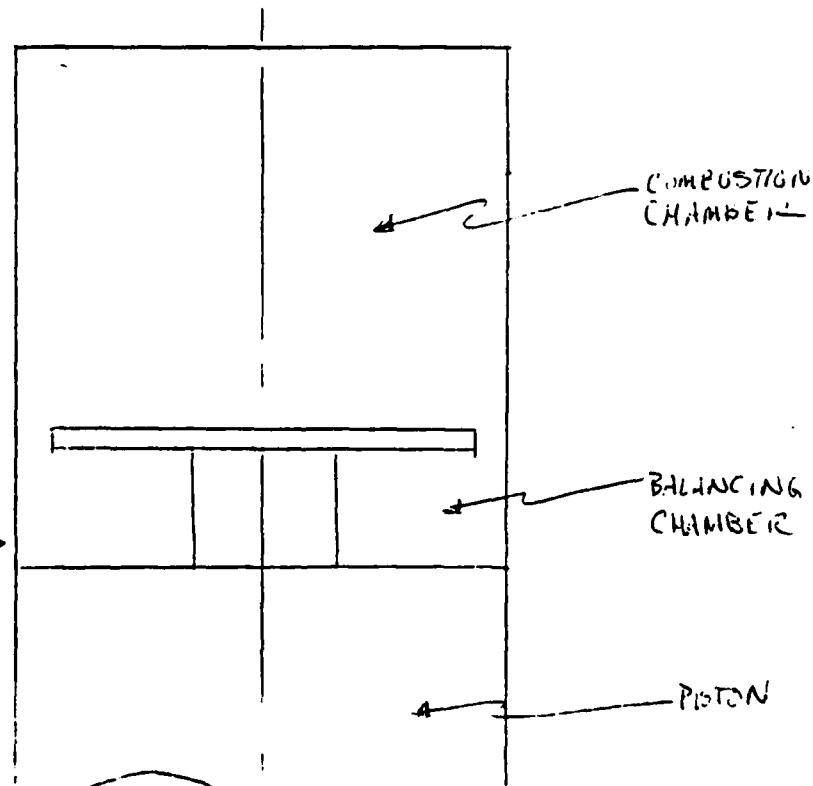
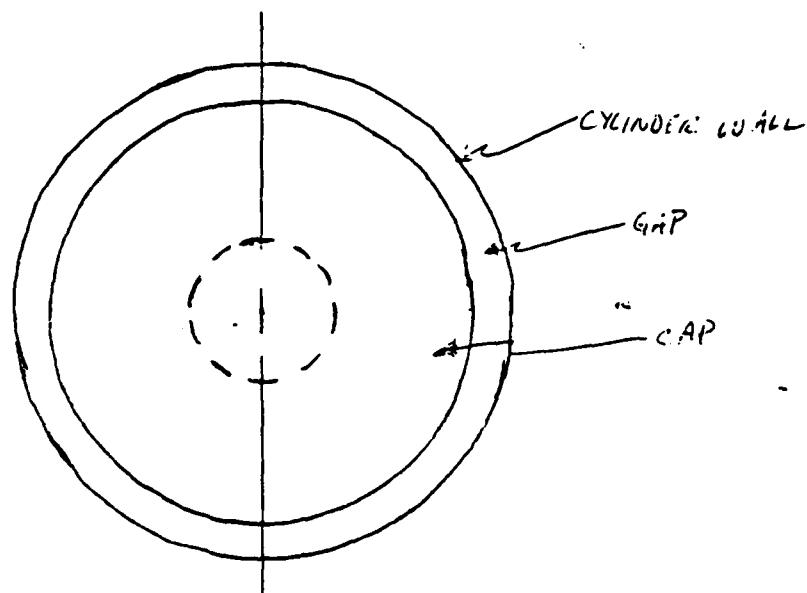
f) equilibrium and quasi-equilibrium cycles

Figure 1 QEHBC THERMODYNAMIC P-V DIAGRAM

TABLE I. QEHB THERMODYNAMIC CYCLE*

| <u>State</u> | | <u>Process</u> | |
|----------------|---------------|---|---------------------------|
| <u>Initial</u> | <u>Final</u> | <u>Region I</u> | <u>Region II</u> |
| 1_I | \rightarrow | compression process isentropic | ----- |
| 1_{II} | \rightarrow | | isentropic |
| 2_I | \rightarrow | first heat addition constant TOTAL volume heat addition | ----- |
| 2_{II} | \rightarrow | | ----- |
| 3_{II} | \rightarrow | second heat addition constant pressure heat addition | isentropic compression |
| 3_I | \rightarrow | | ----- |
| 4_{II} | \rightarrow | expansion process isentropic | ----- |
| 4_I | \rightarrow | | ----- |
| 5_{II} | \rightarrow | heat rejection constant volume | isentropic |
| 5_I | \rightarrow | | ----- |
| 1_{II} | \rightarrow | constant volume | constant volume |
| 1_I | \rightarrow | | ----- |

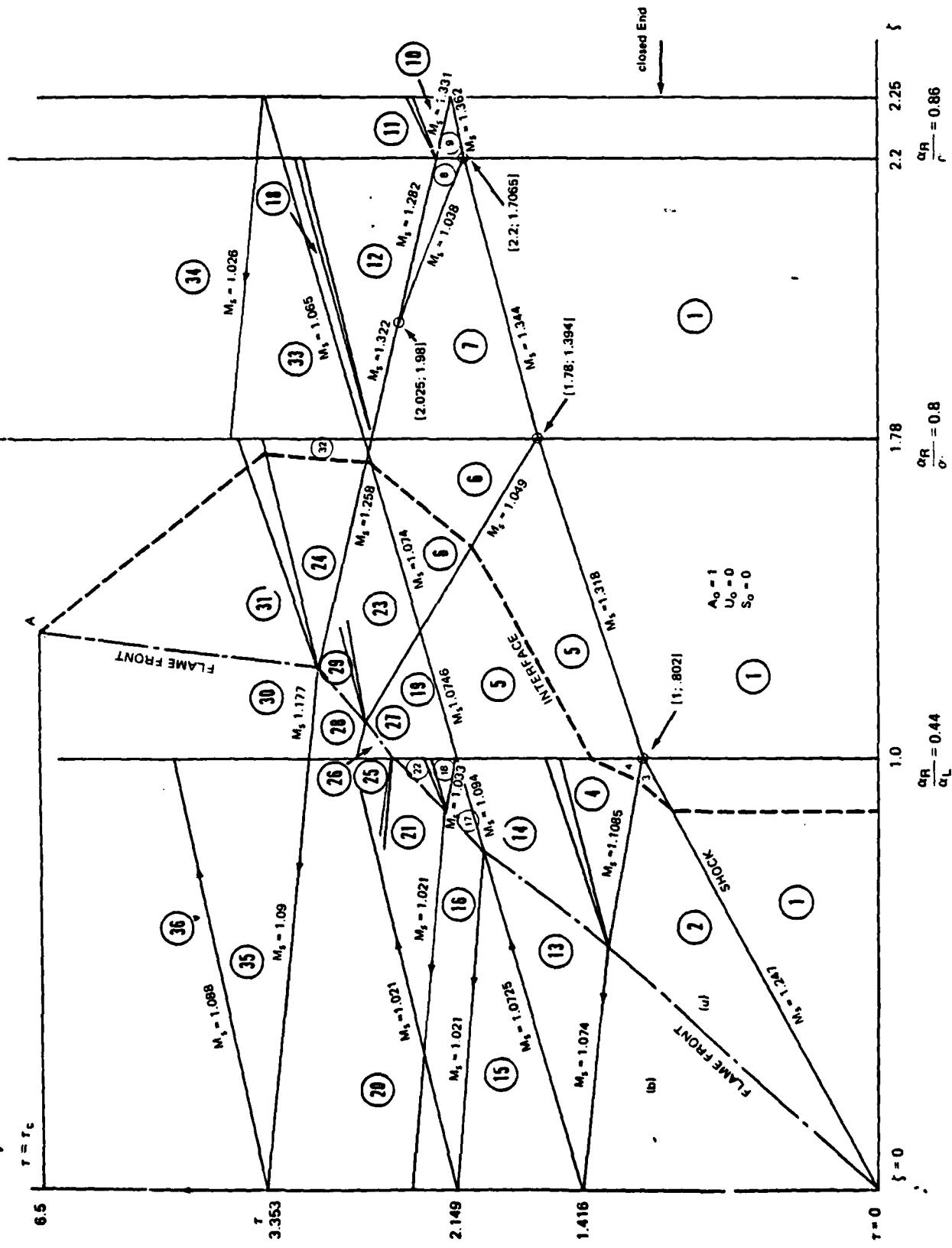
*To attain the QEHB cycle, regions I and II must remain separated throughout the cycle and all processes are quasi-equilibrium in nature. If the two regions mix and attain equilibrium the cycle produces the classical combined or dual cycle.



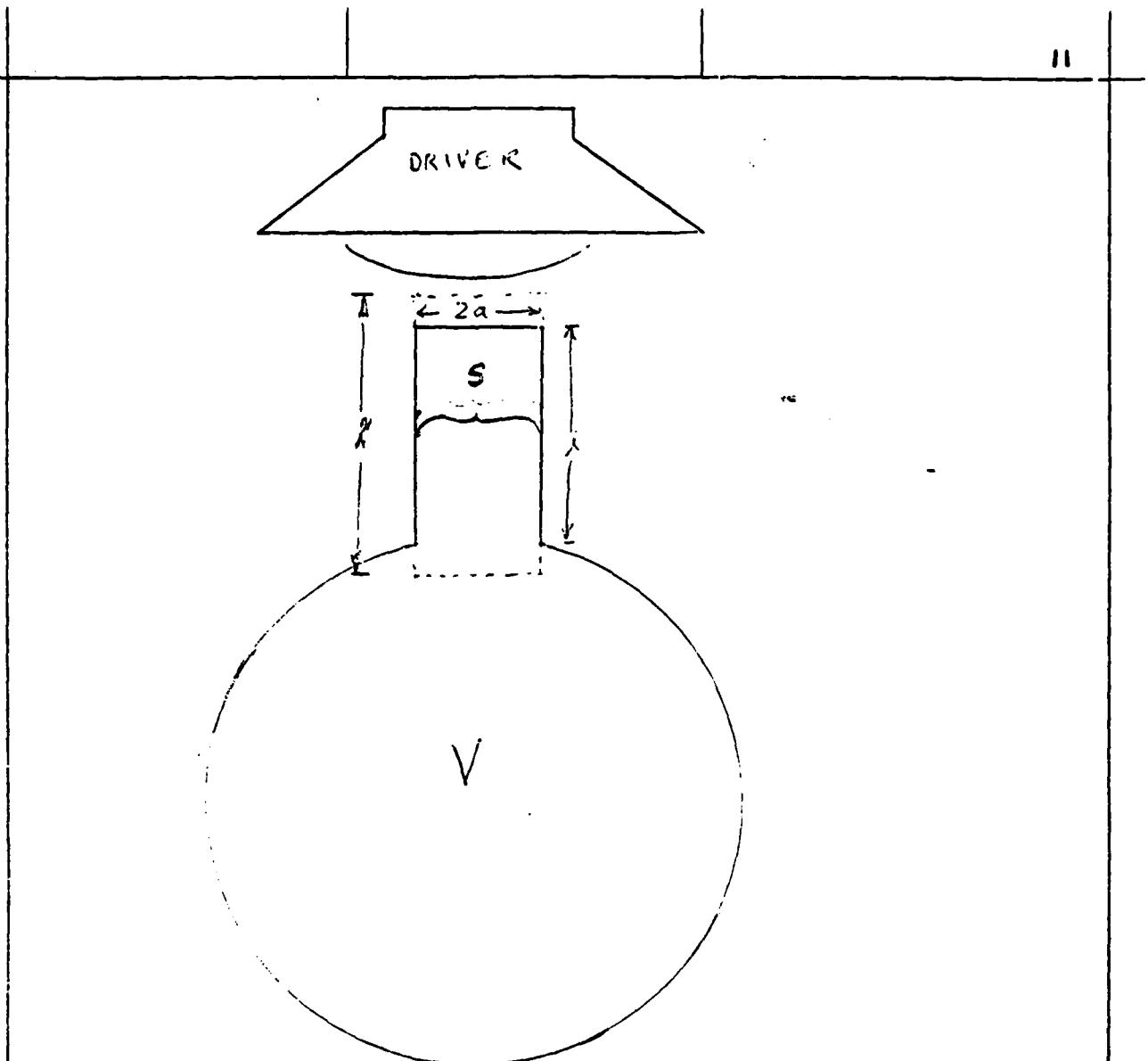
THE NAHBE PISTON
FIGURE 2

NONSTEADY FLOW WAVE DIAGRAM (QUALITATIVE)

FIGURE 3



100 SHEETS
100 SHEETS
100 SHEETS
100 SHEETS



V : Volume

$2a$: Gap opening

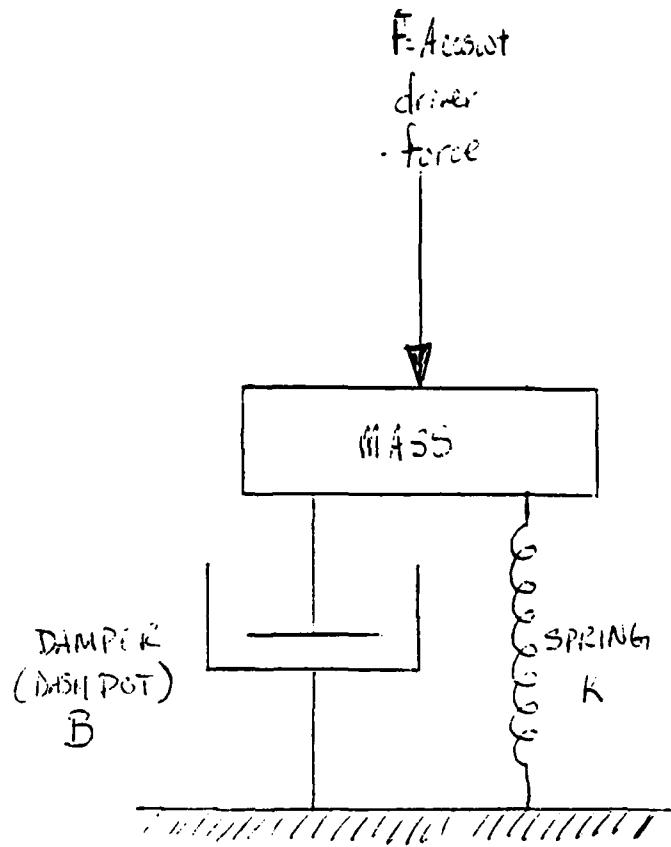
l : Gap length

S : Cross sectional Area of Gap.

A HELM HOLTZ RESONATOR

FIGURE 9

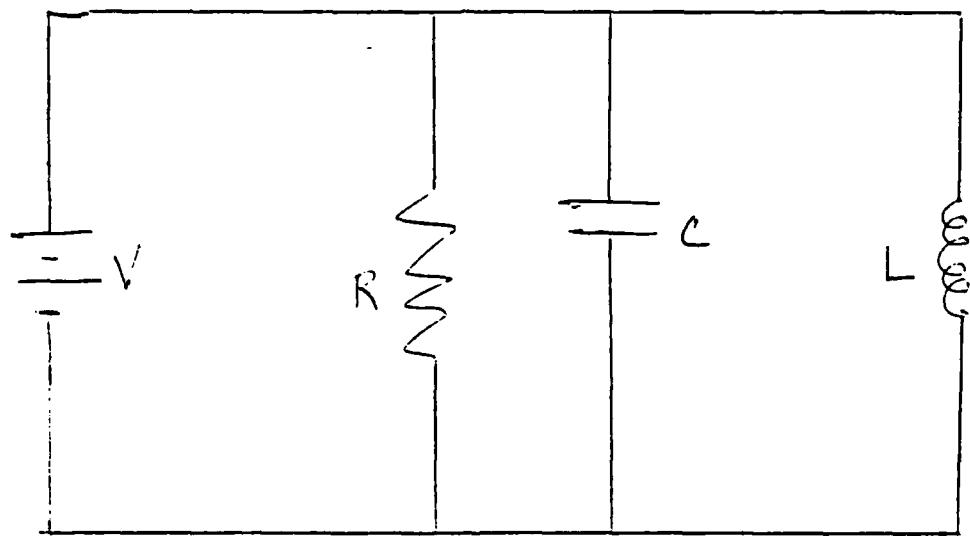
100 SHEETS 100 SHEETS 100 SHEETS
100 SHEETS 100 SHEETS 100 SHEETS



A MECHANICAL MODEL OF
A HELMHOLTZ RESONATOR

FIGURE 5

NATIONAL | 43-331 10 SHEETS 1 QUARTER
43-332 100 SHEETS 1 QUARTER
43-333 200 SHEETS 1 QUARTER



ELECTRICAL ANALOG
FOR
A HELMHOLTZ RESONATOR

FIGURE 6

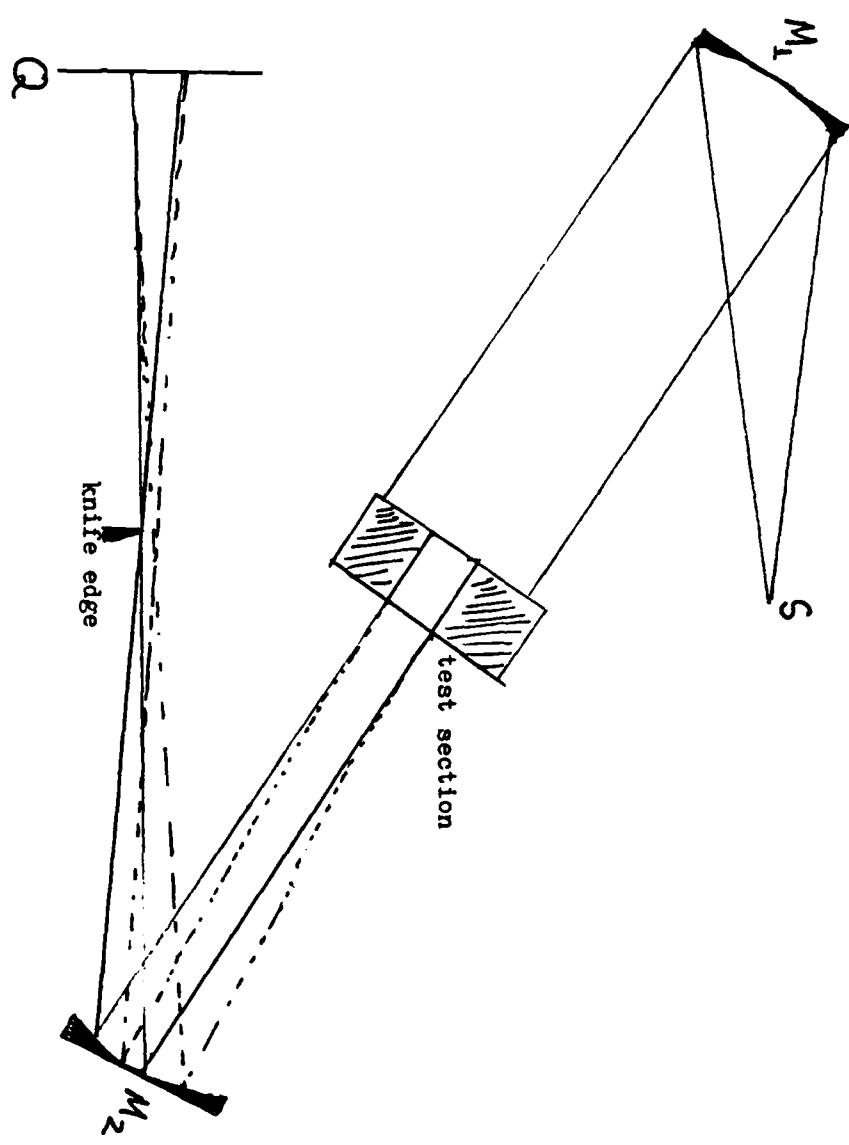


Figure 7

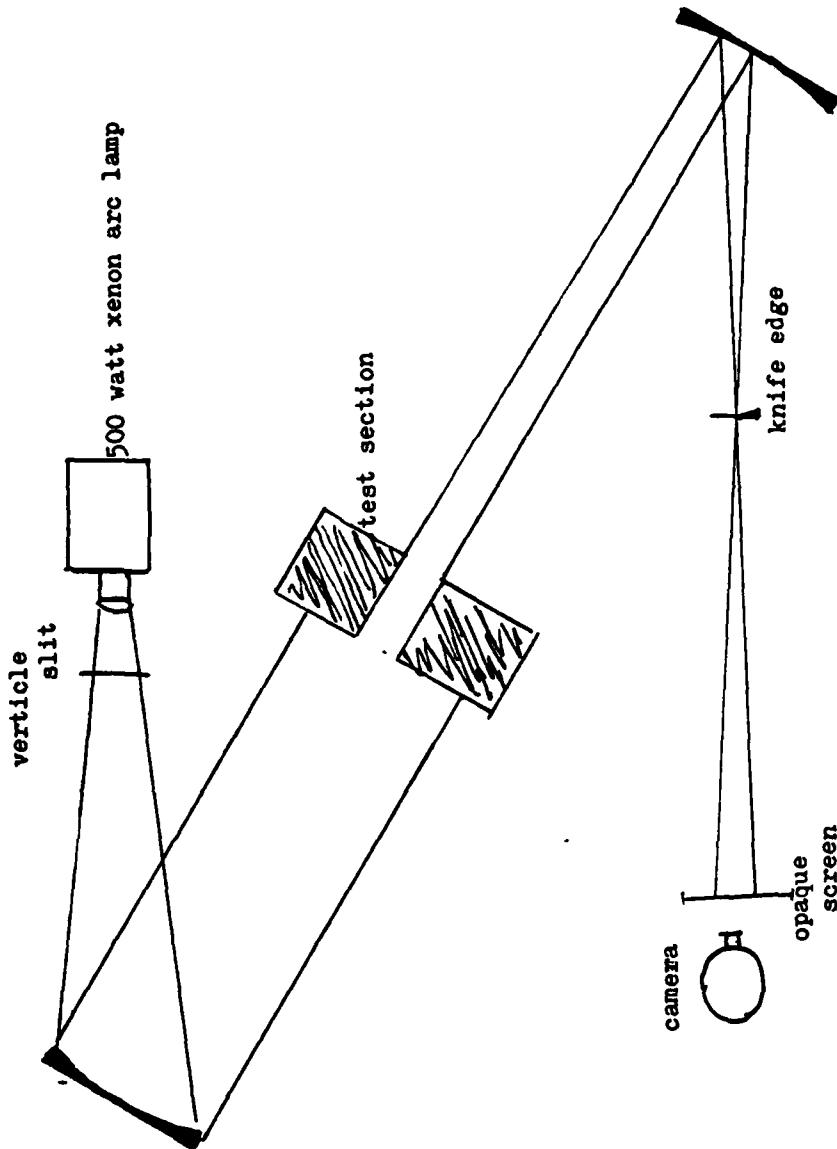


Figure 8

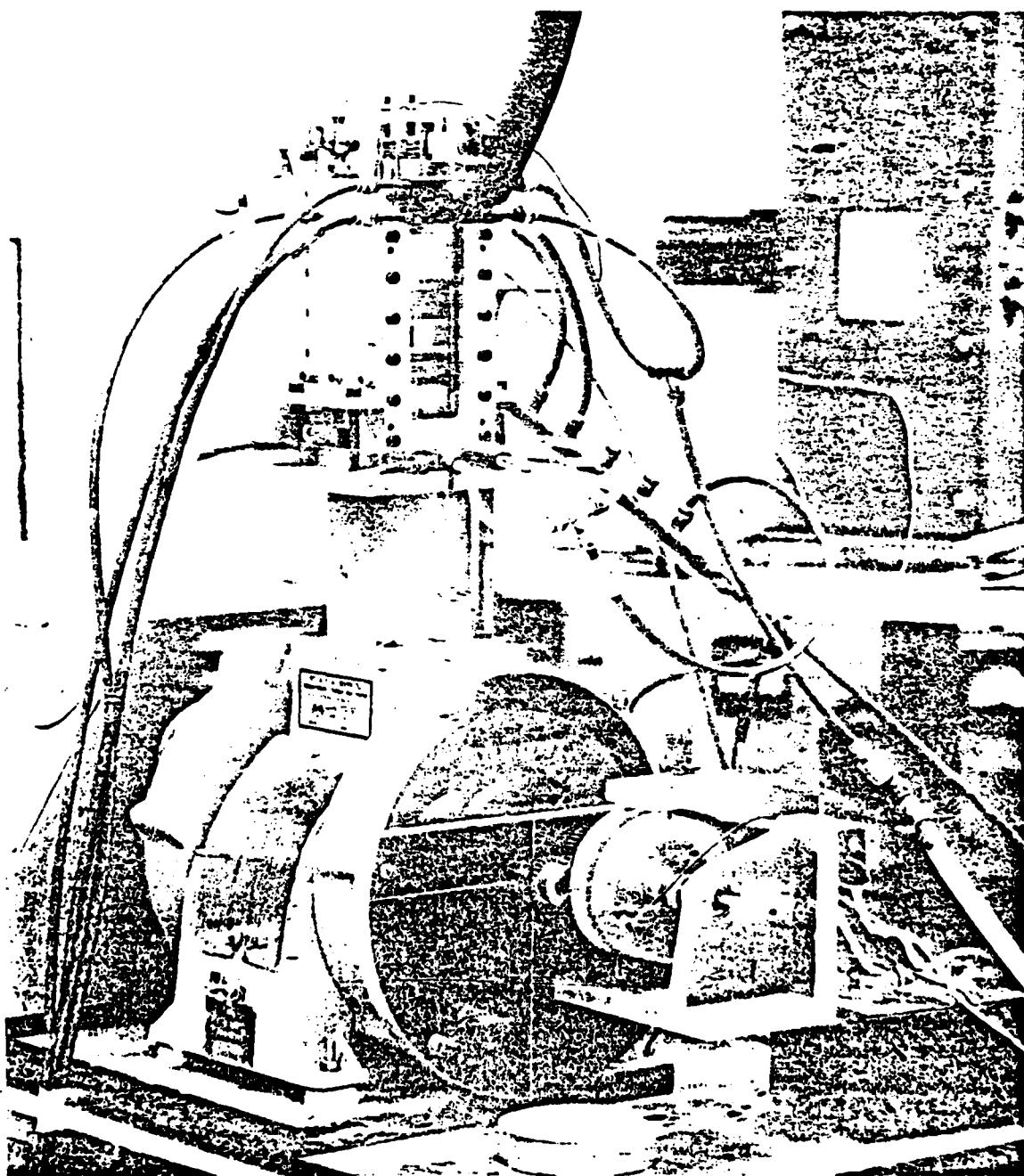


FIGURE 9

Overall View of 2-Dimensional Square Engine

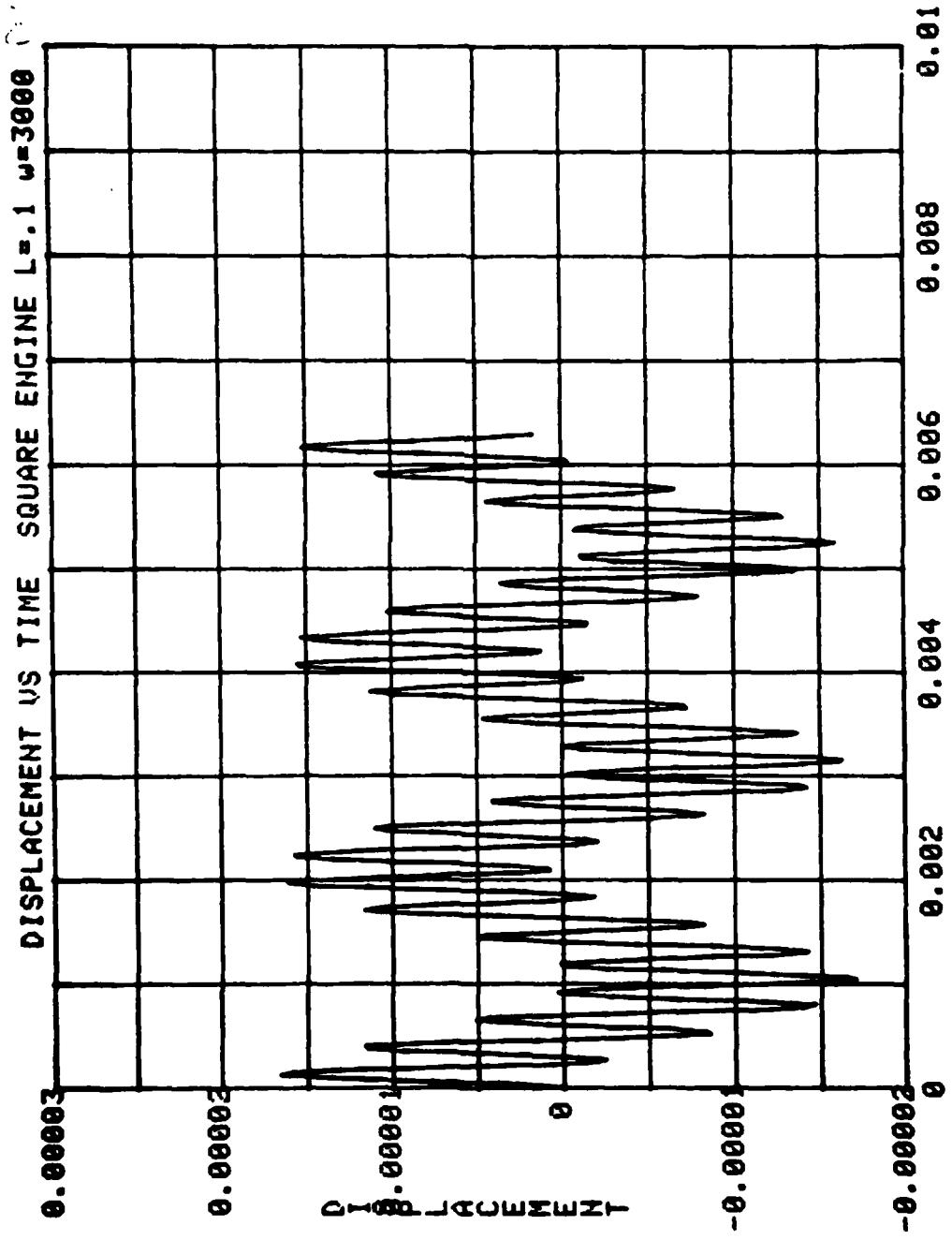
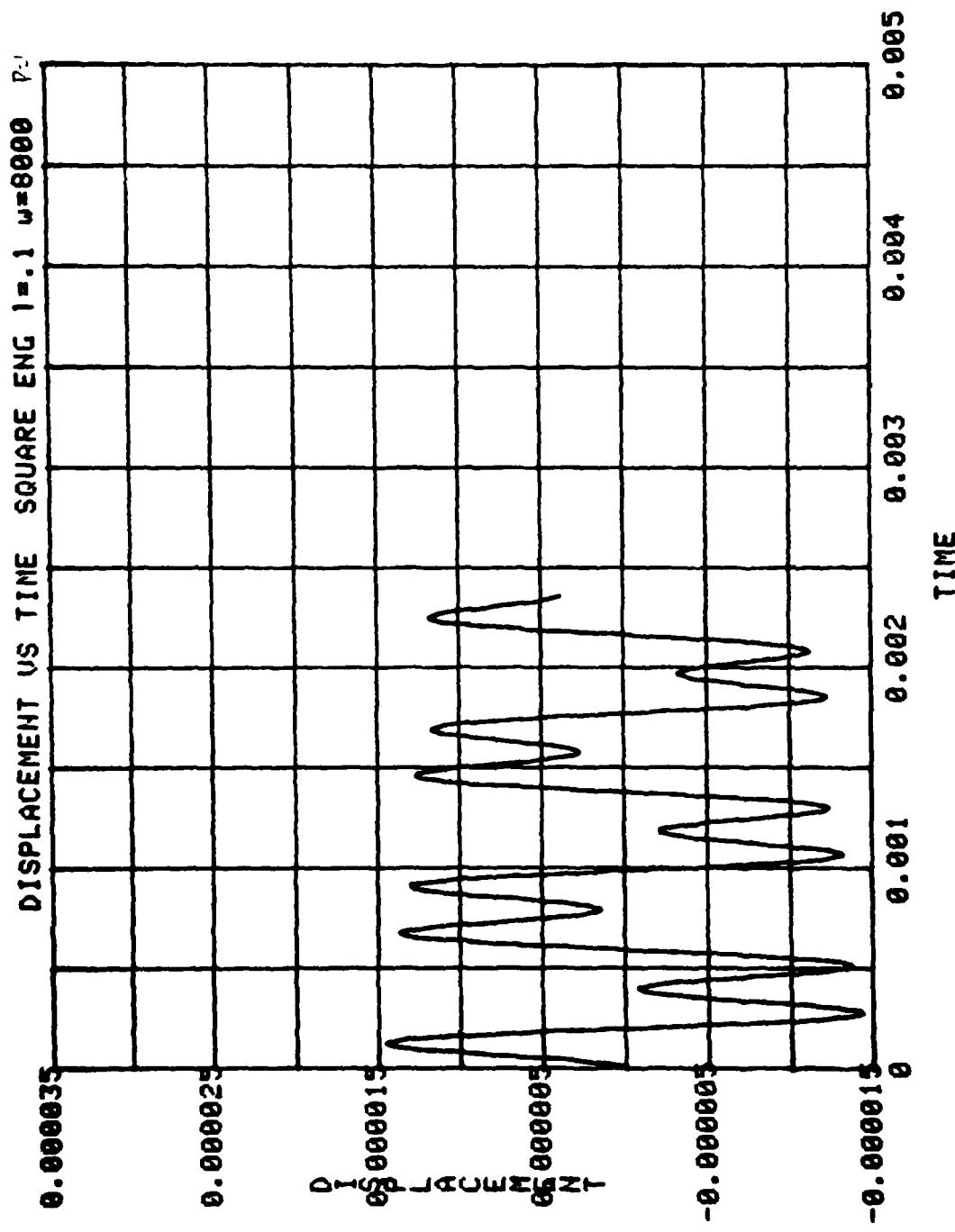


FIGURE 10

FIGURE 11



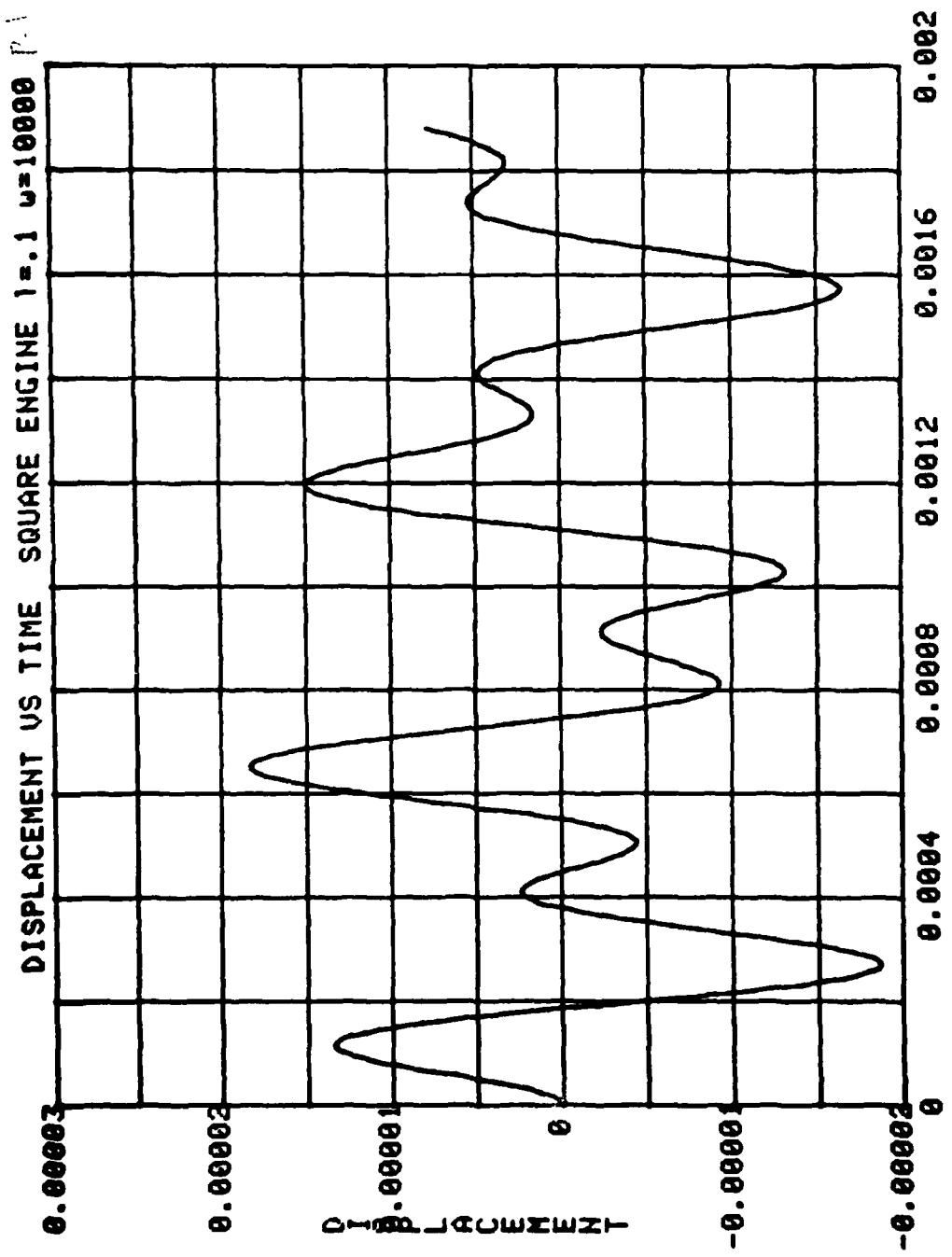
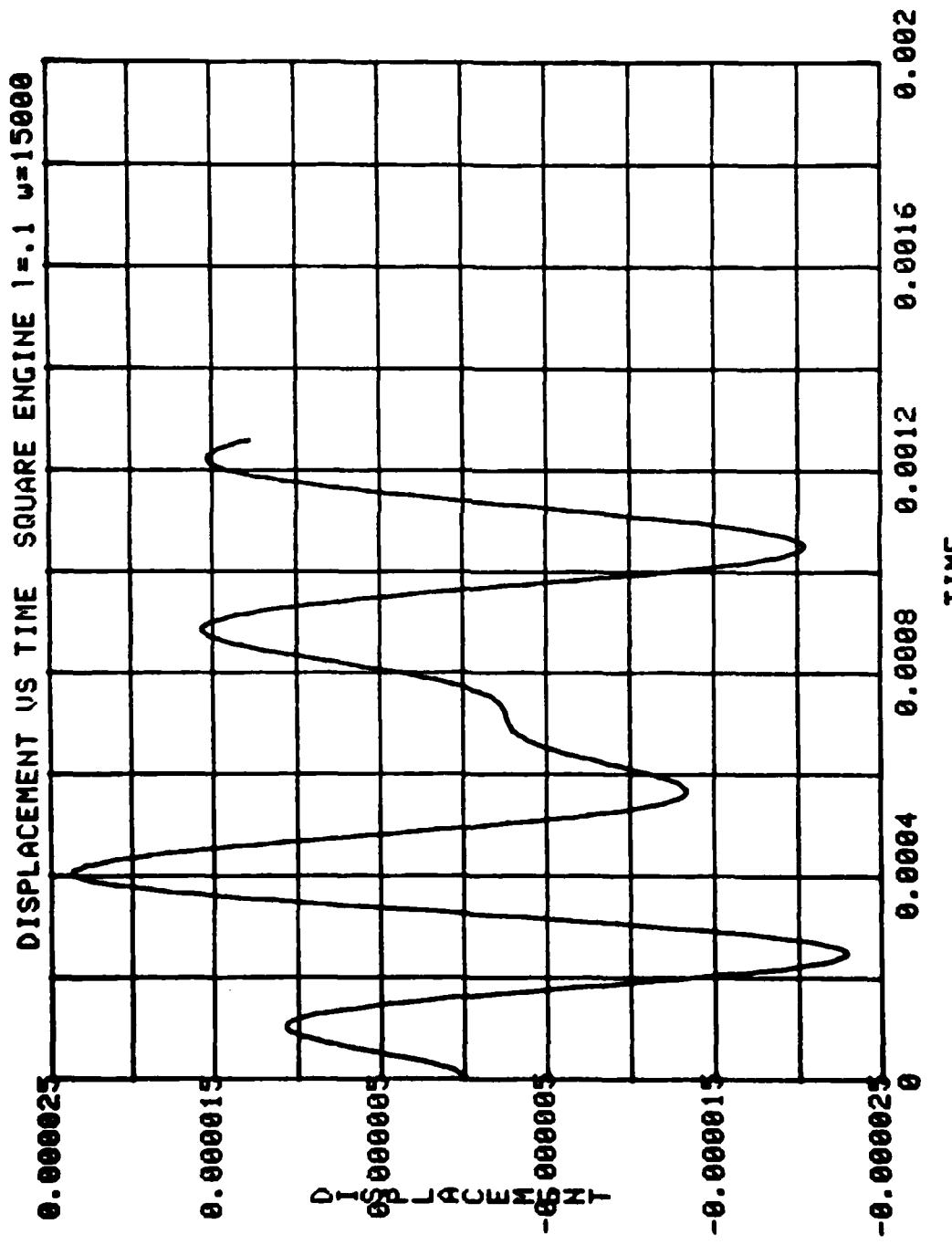


FIGURE 12

FIGURE 13



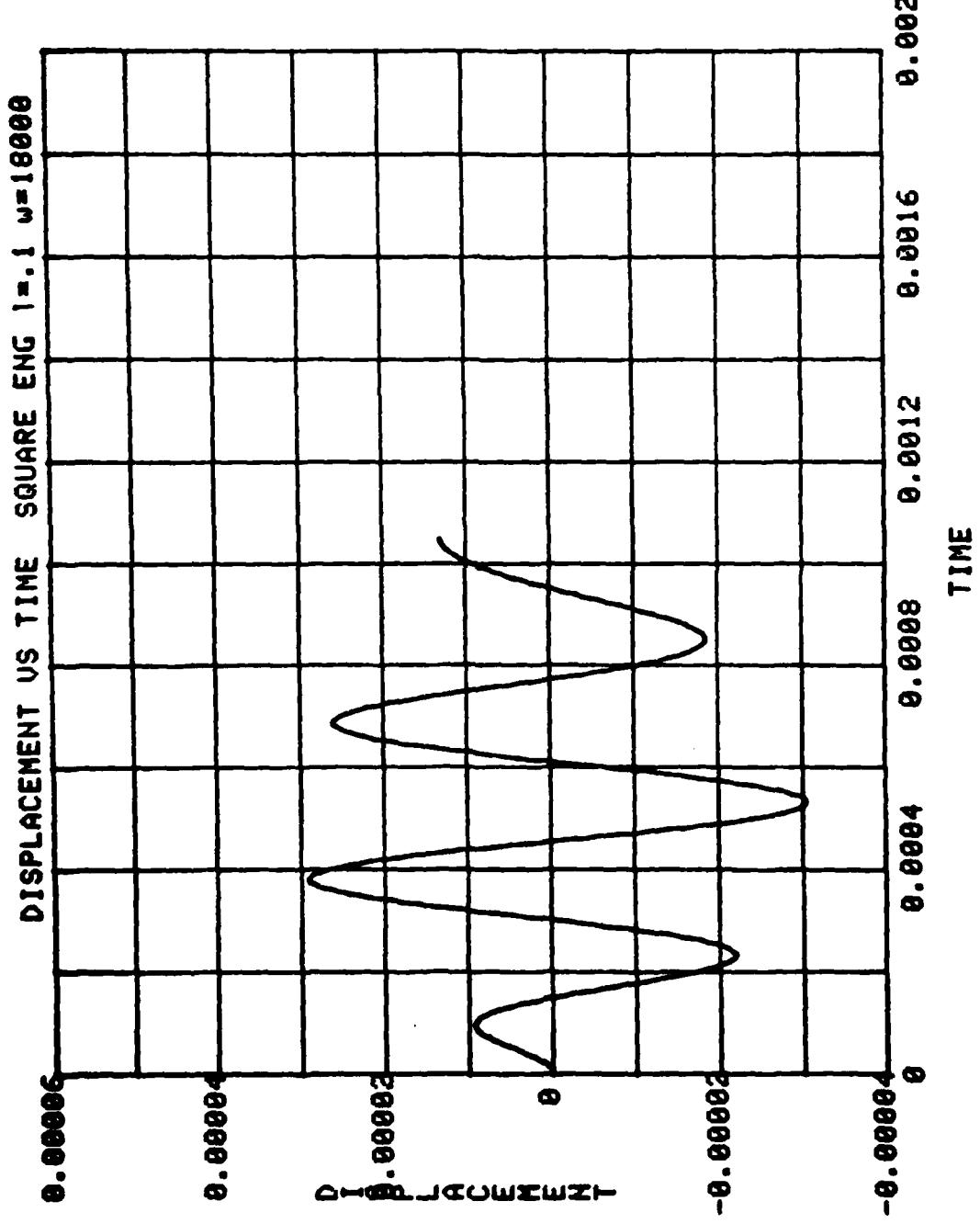


FIGURE 14

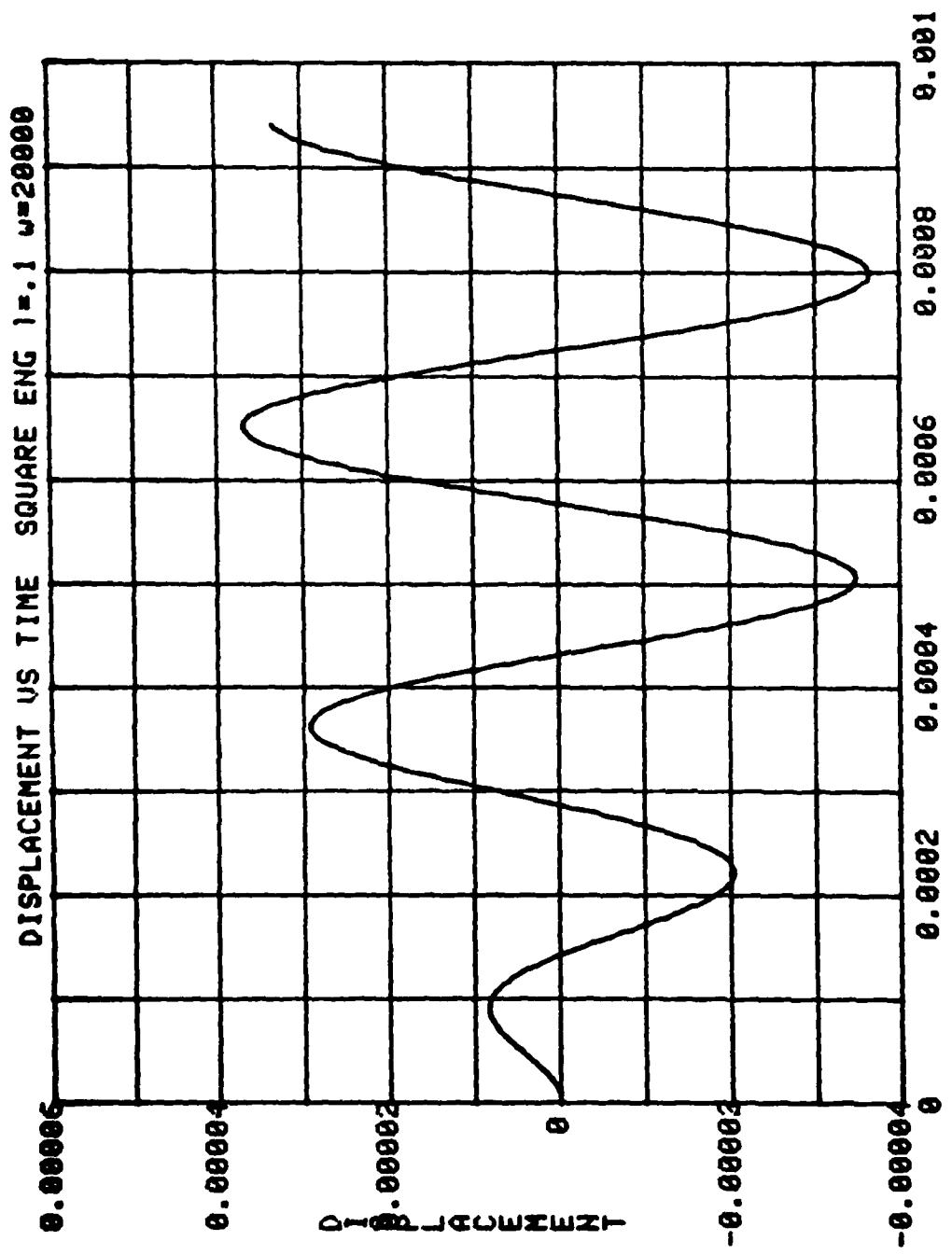


FIGURE 15

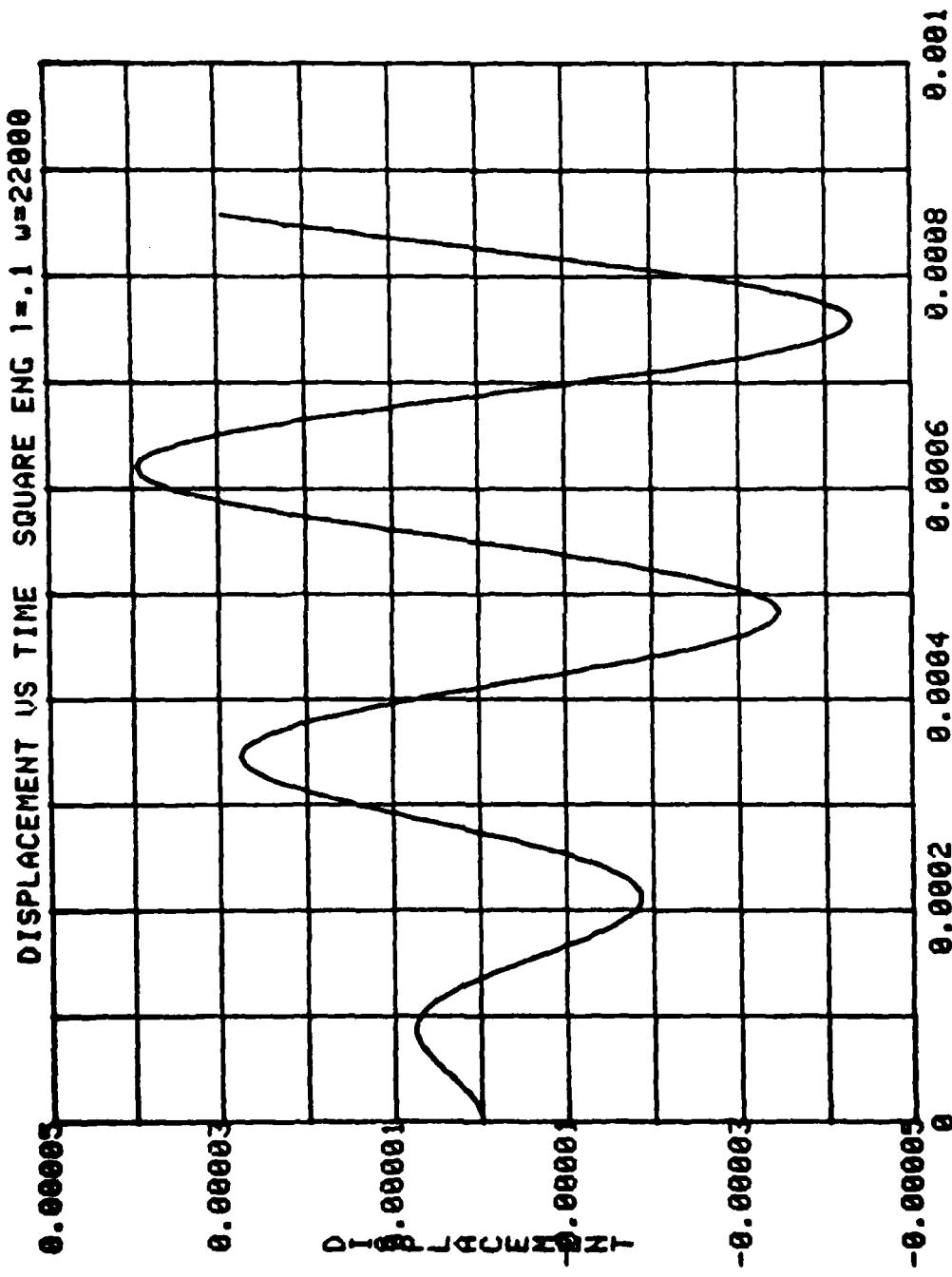


FIGURE 16

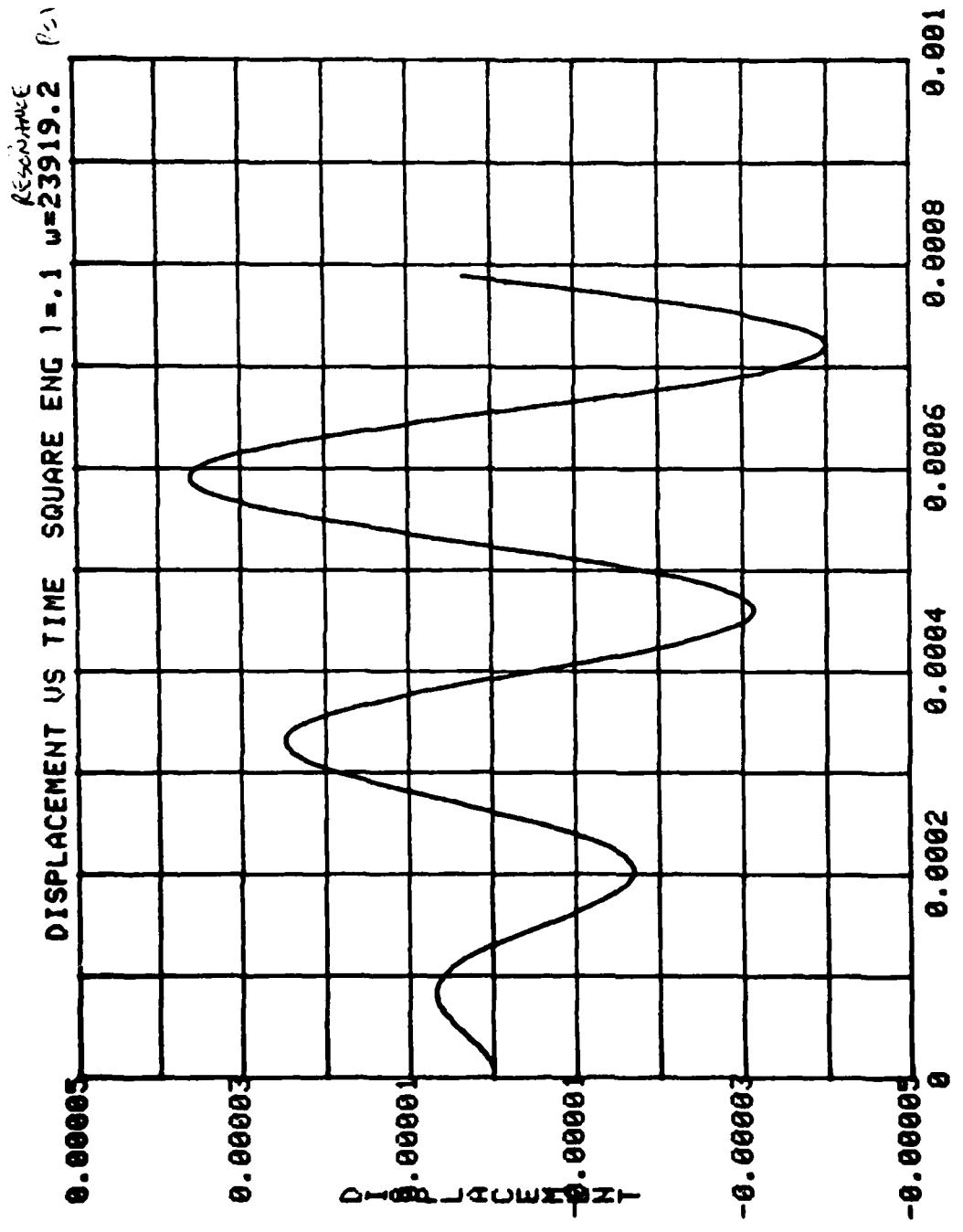
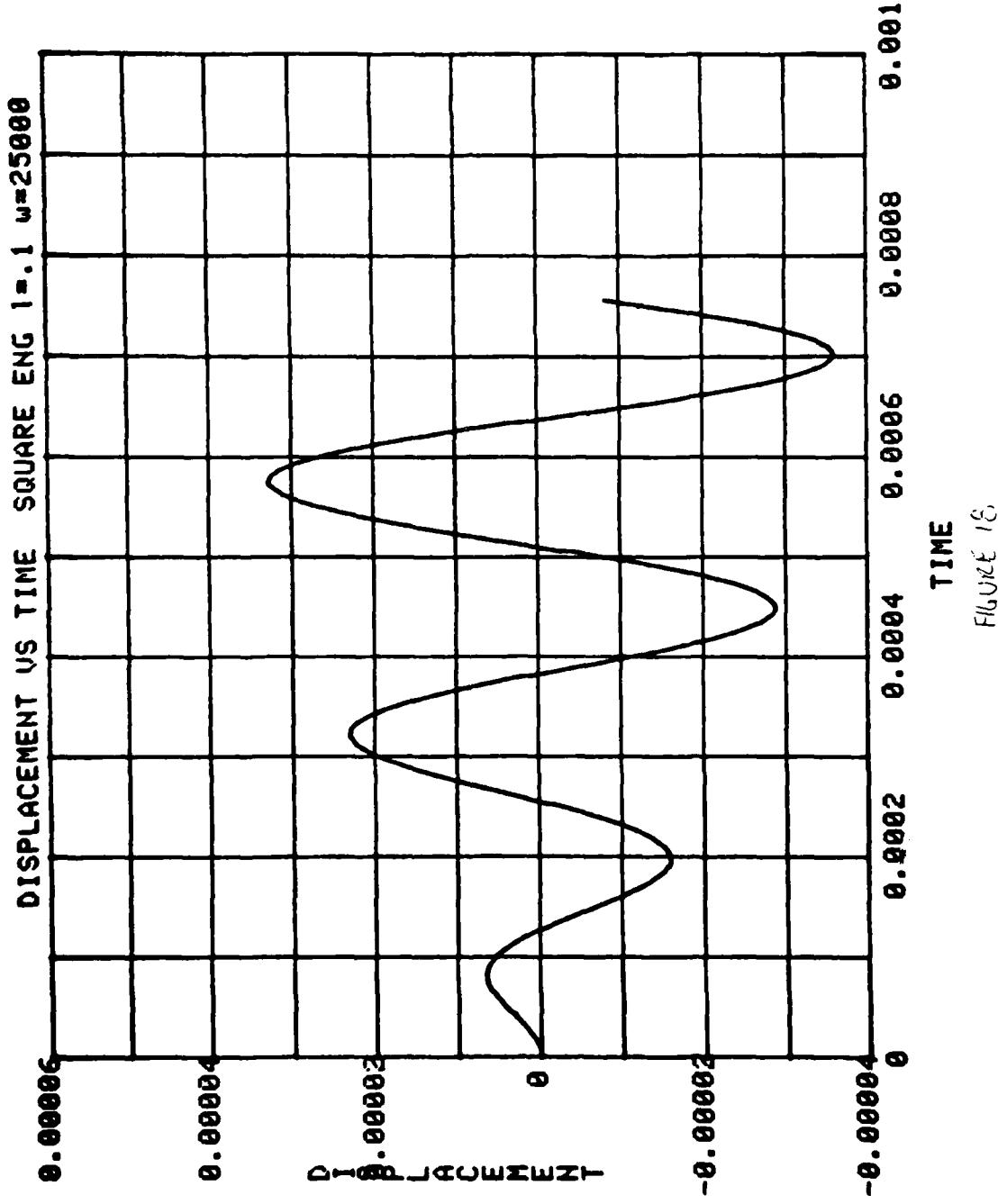


Figure 17



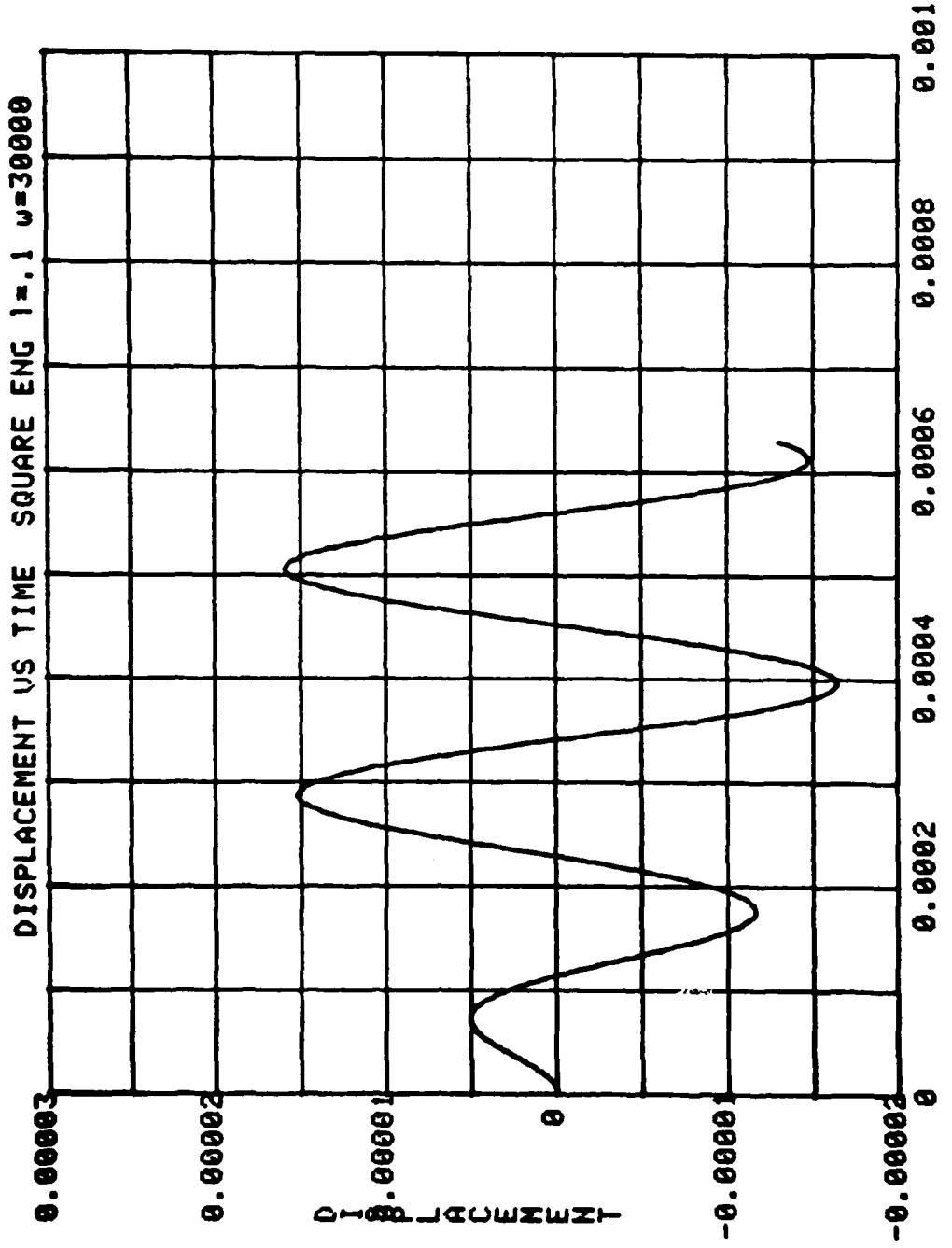
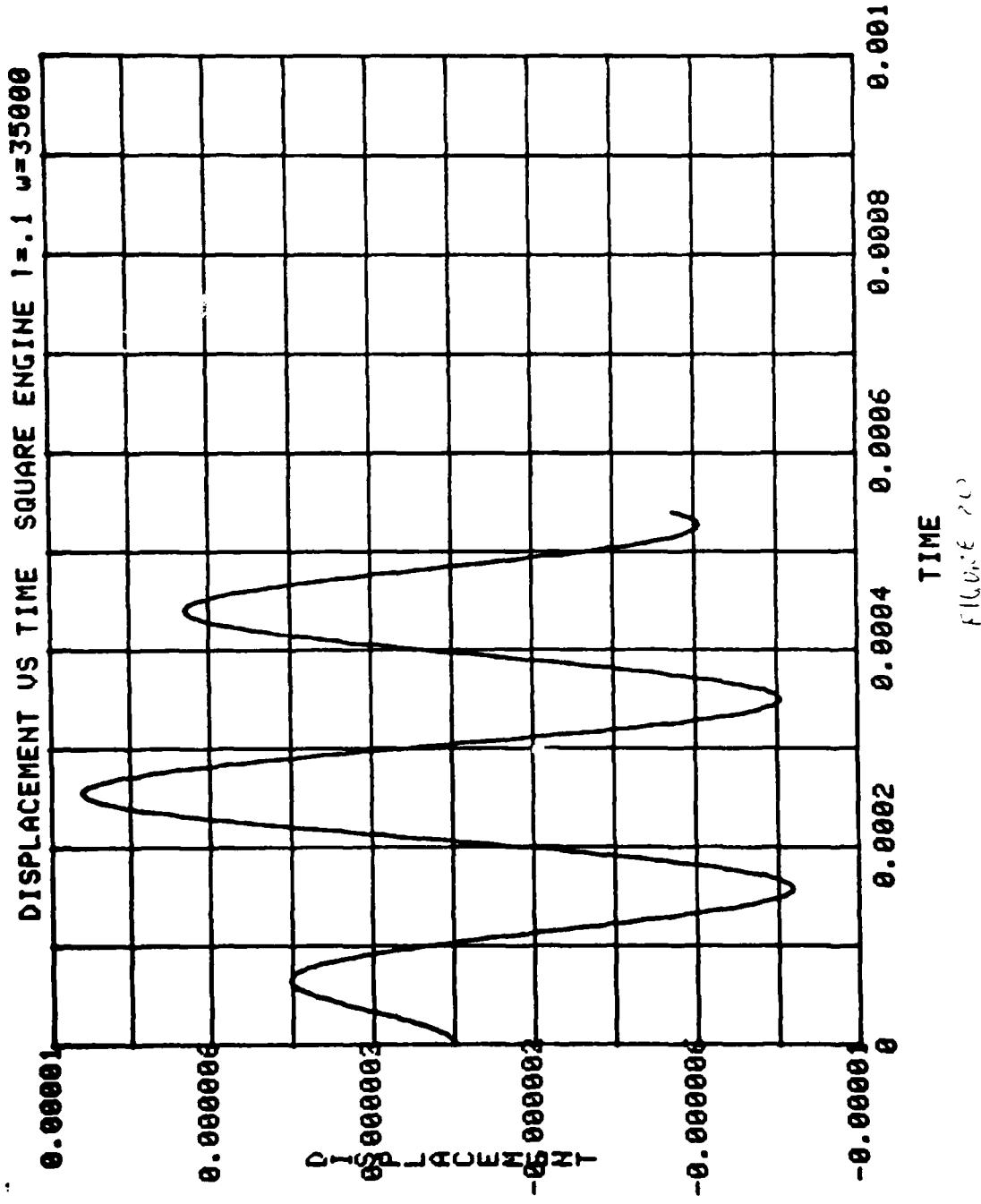
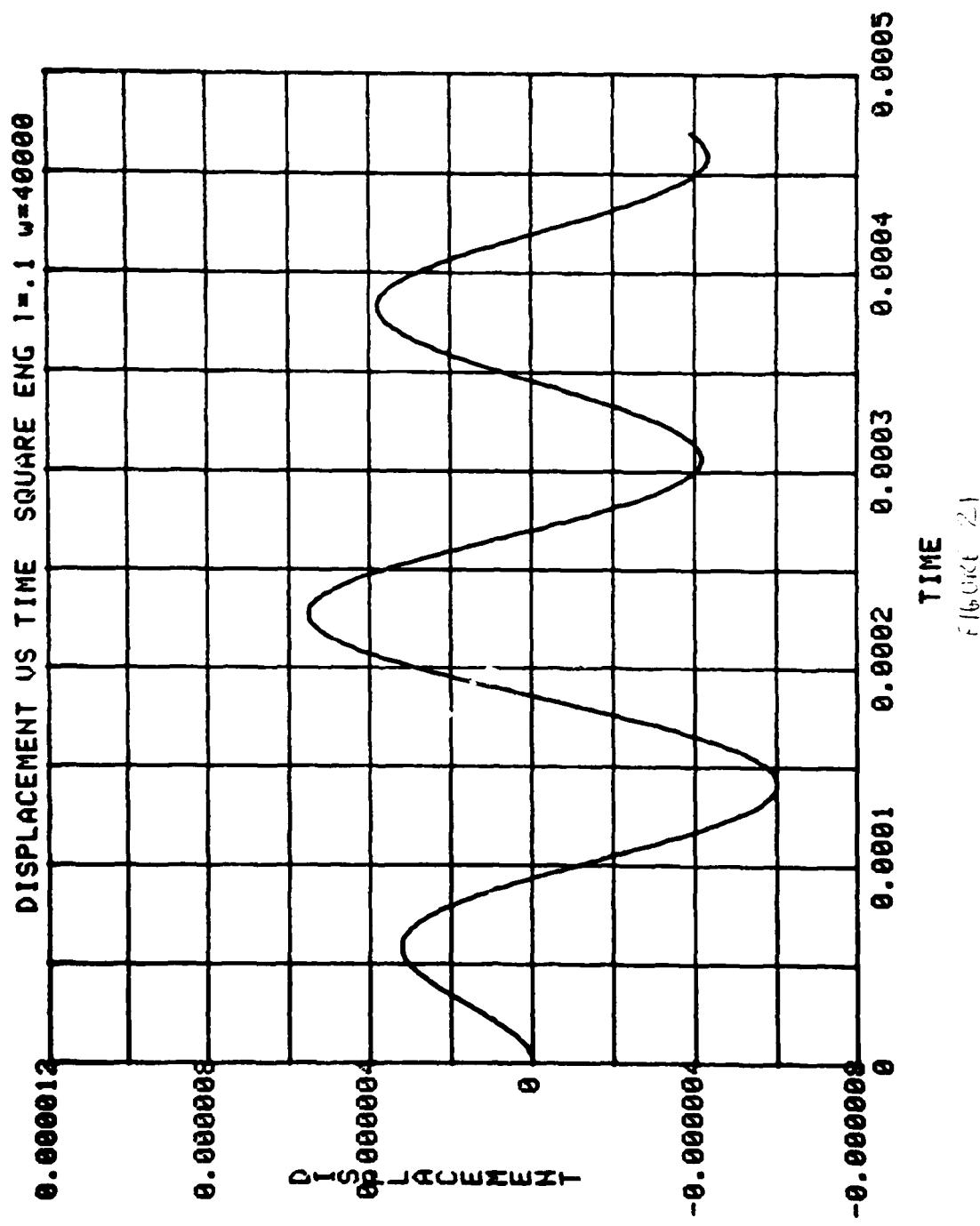


FIGURE 19





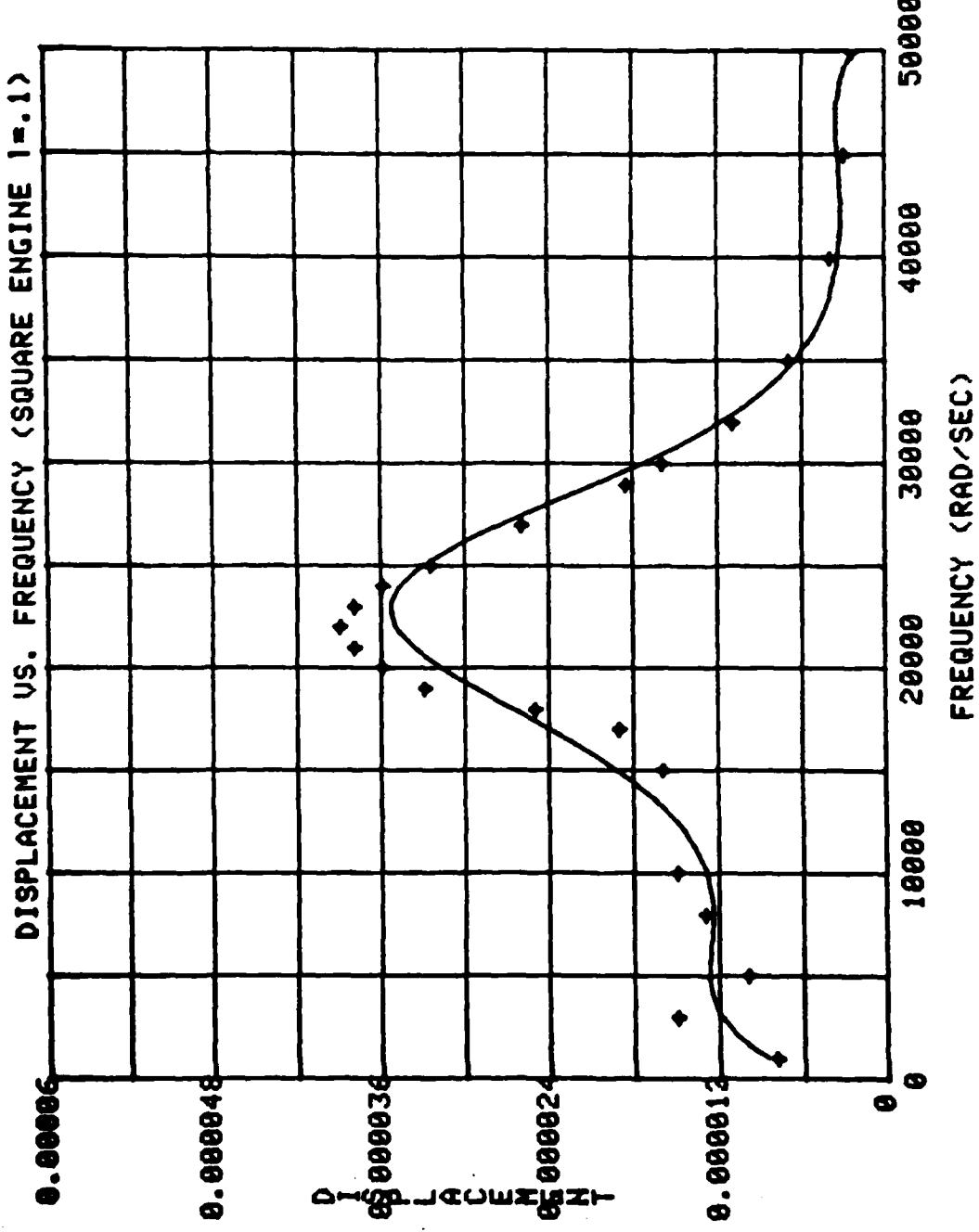


FIGURE 2.2

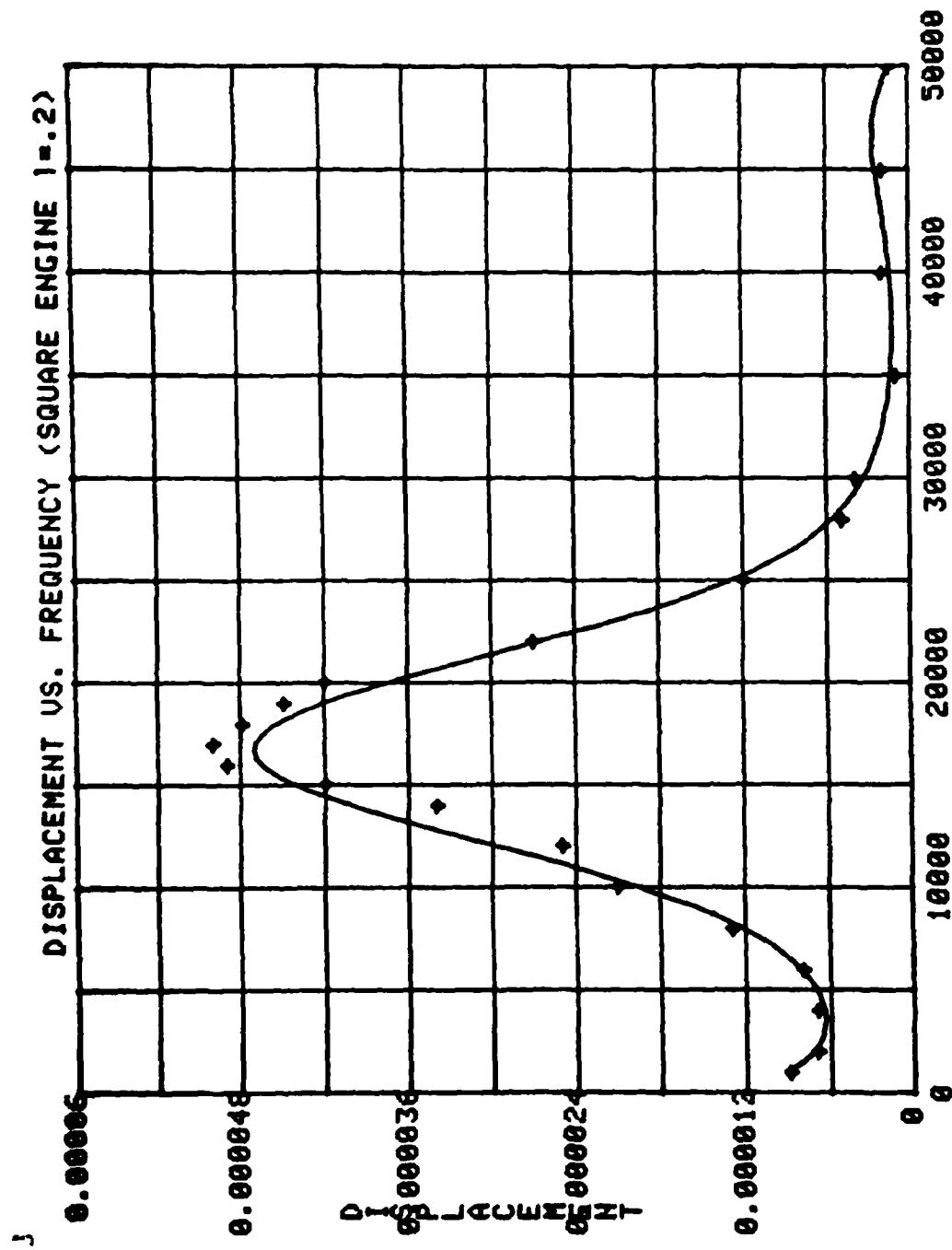


FIGURE 23

5.

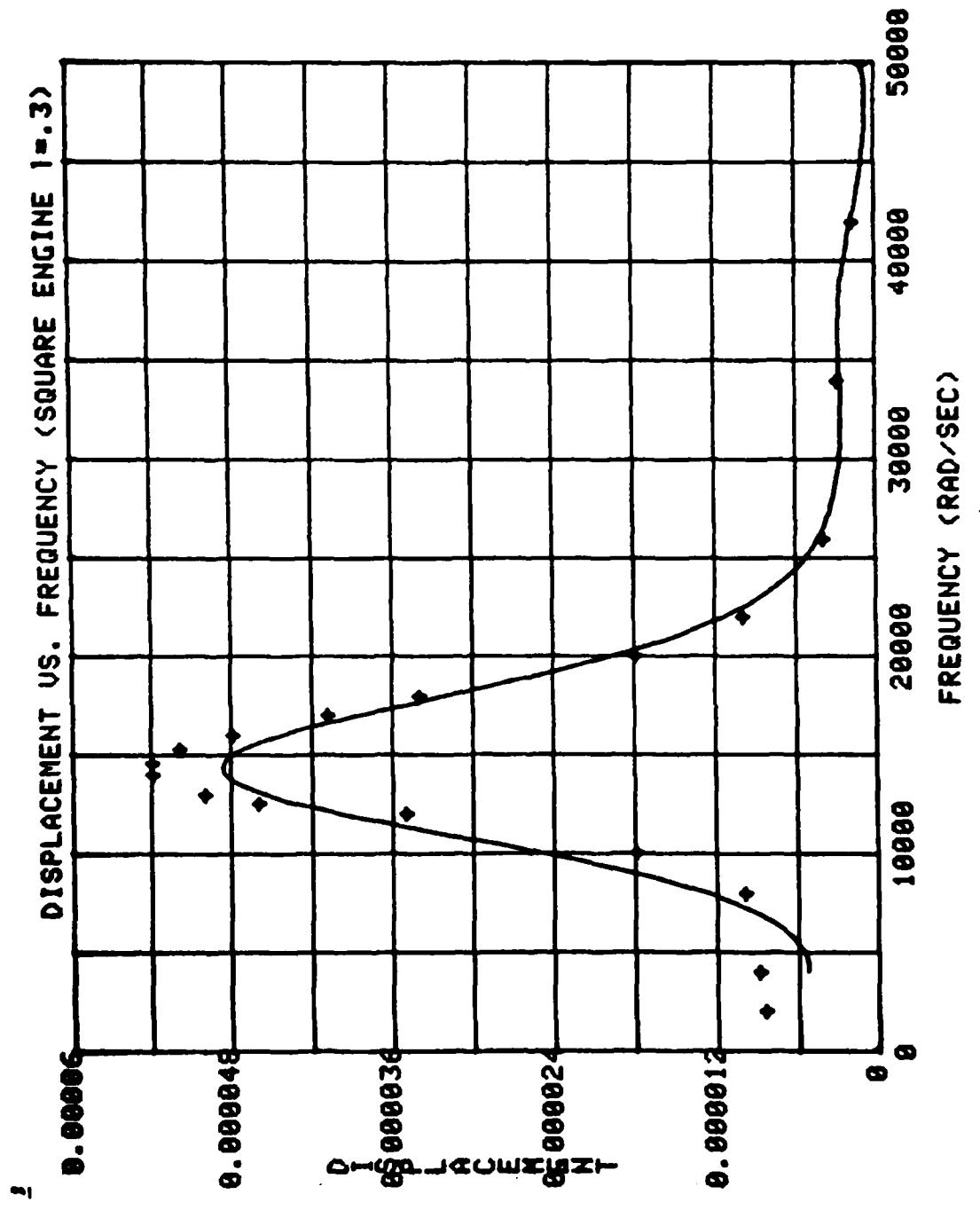


FIGURE 29

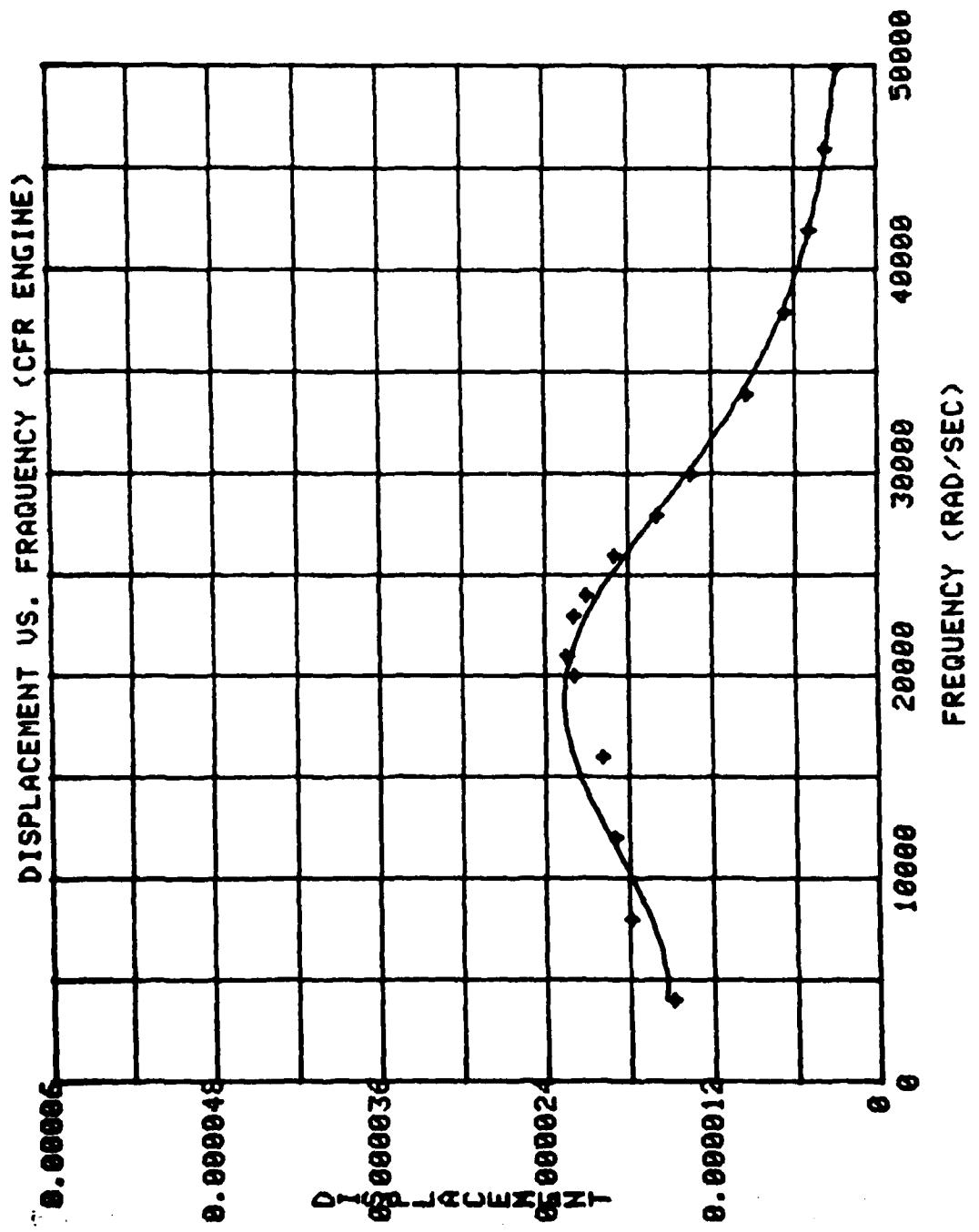


FIGURE 25

Figure 26



Figure 27



Figure 28



Figure 29

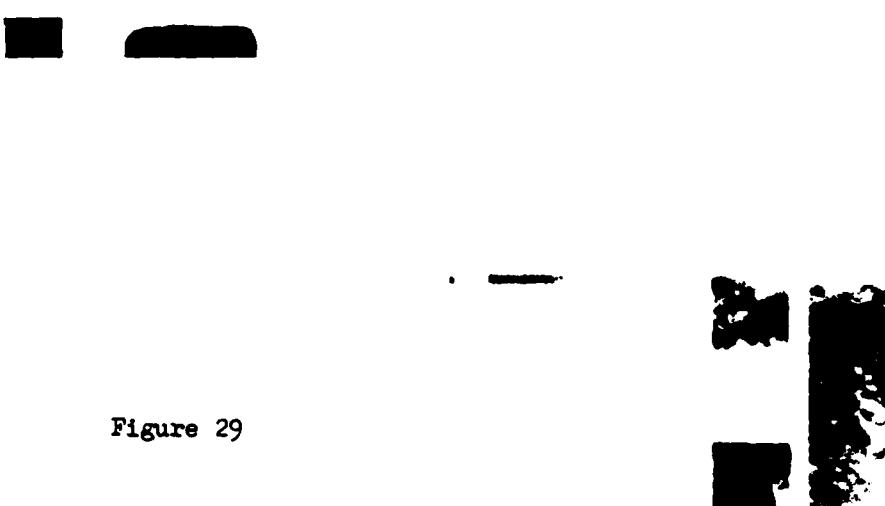


Figure 30

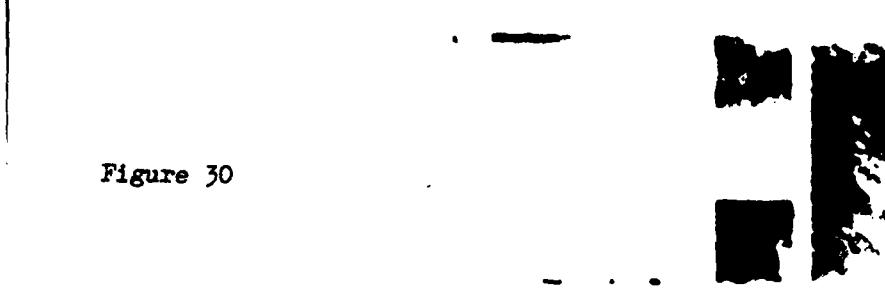


Figure 31

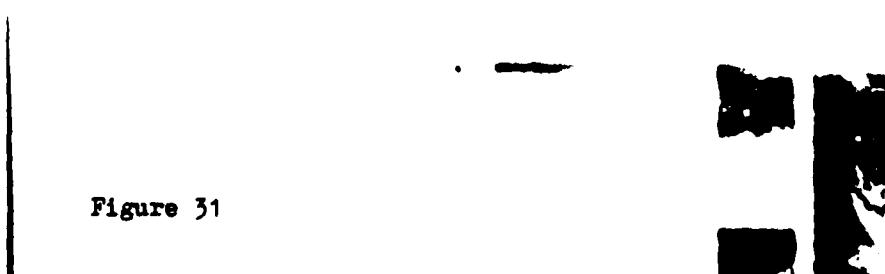


Figure 32



Figure 33



Figure 34



Figure 35

NOTE
RINGS



Figure 36

NOTE
RINGS



Figure 37



Figure 38



Figure 39



Figure 40



Figure 41



Figure 42



Figure 43



Figure 44



Figure 45

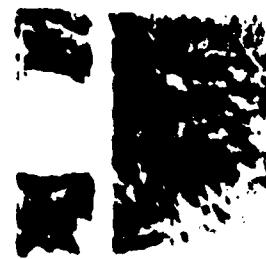


Figure 46

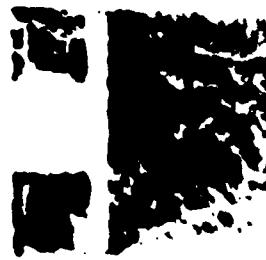


Figure 47



Figure 48



Figure 49



Figure 50

SECONDARY COMBUSTION BEGINNING



Figure 51

NOTE RINGS



MORE PLUNGE NEAR CAP LIP

Figure 52

SECONDARY COMBUSTION FULL FLEDGED



Figure 53



Figure 54

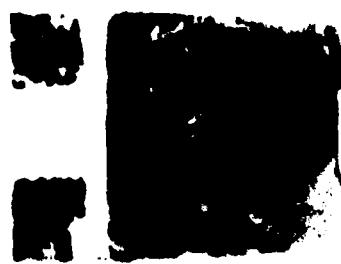


Figure 55



Figure 56

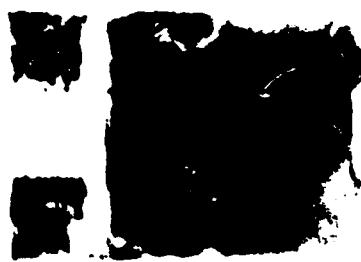


Figure 57



Figure 58



Figure 59



Figure 60

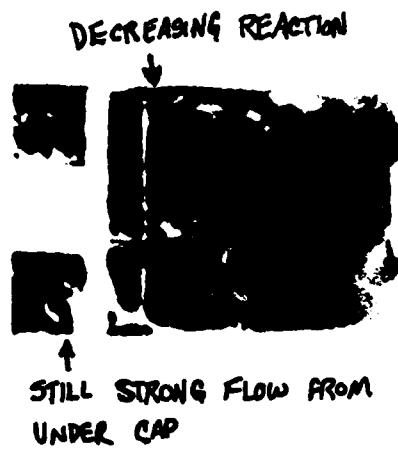


Figure 61



Figure 62

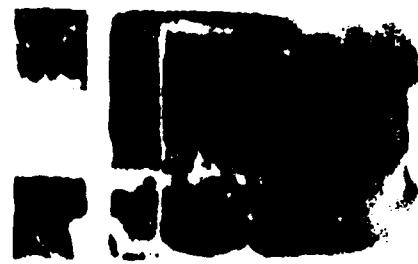


Figure 63

COMBUSTION QUENCHED

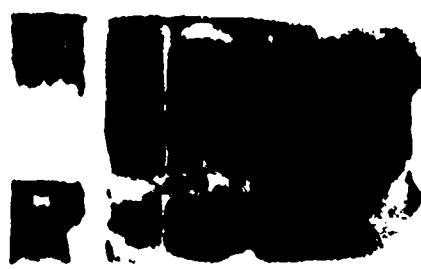


Figure 64



LIS

FREQDATA 30 Mar 81 14:32

| | | |
|---------|---|-----|
| 15292.9 | , | 0.6 |
| 13198.2 | , | 0.7 |
| 11469.7 | , | 0.8 |
| 19195.3 | , | 0.9 |
| 9175.76 | , | 1.1 |
| 8341.6 | , | 1.1 |
| 7646.47 | , | 1.2 |
| 7958.28 | , | 1.3 |
| 6554.12 | , | 1.4 |
| 6117.18 | , | 1.5 |
| 5734.85 | , | 1.6 |
| 5397.51 | , | 1.7 |
| 5997.65 | , | 1.8 |
| 4829.35 | , | 1.9 |
| 4587.88 | , | 2. |
| 4369.41 | , | 2.1 |
| 4178.8 | , | 2.2 |
| 3989.46 | , | 2.3 |
| 3823.24 | , | 2.4 |
| 3670.31 | , | 2.5 |
| Ready | | |

Figure 65

INIZIAR SPEDIZIONE

FOR NO TEMPERATURE VARIATION DURING CYCLE

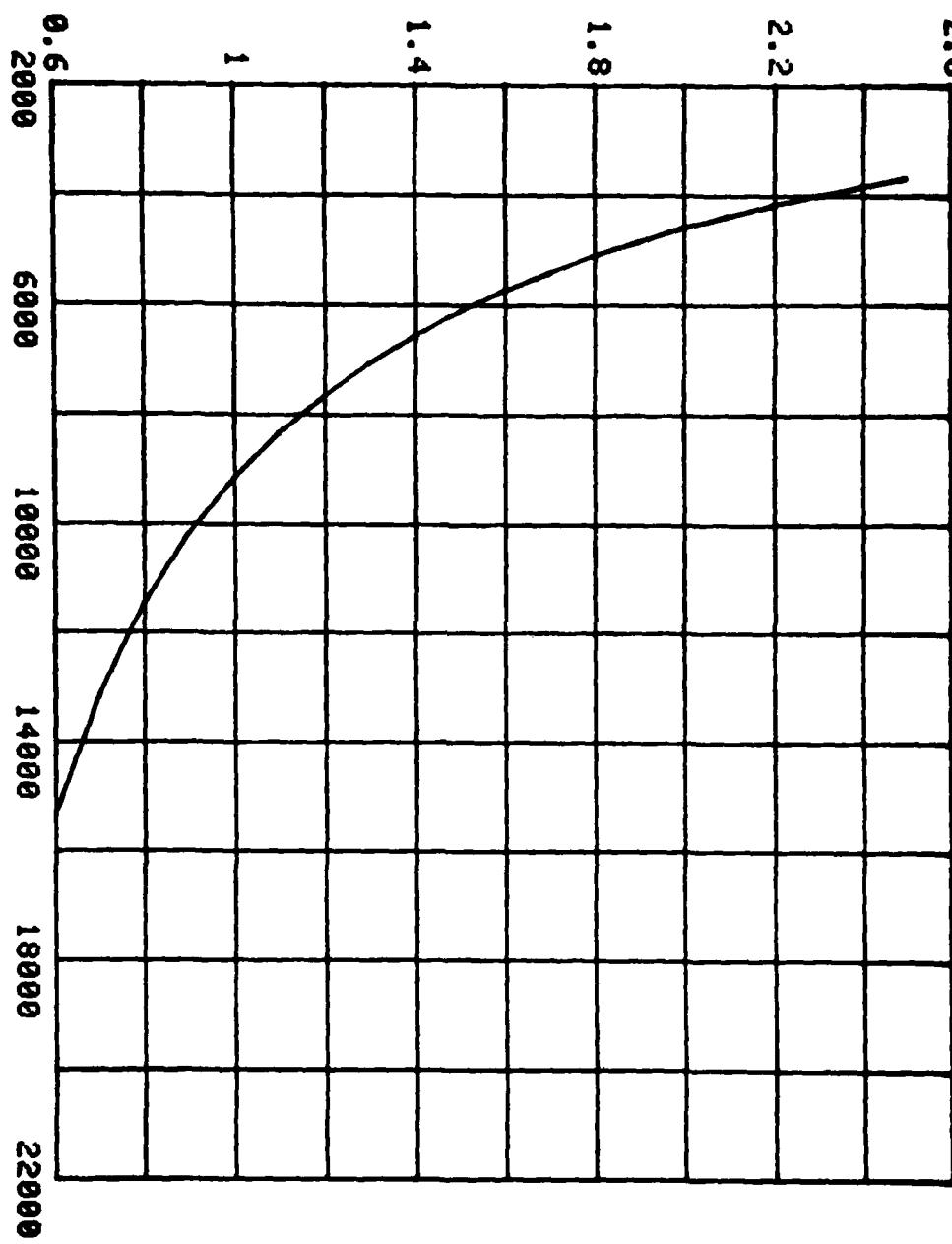
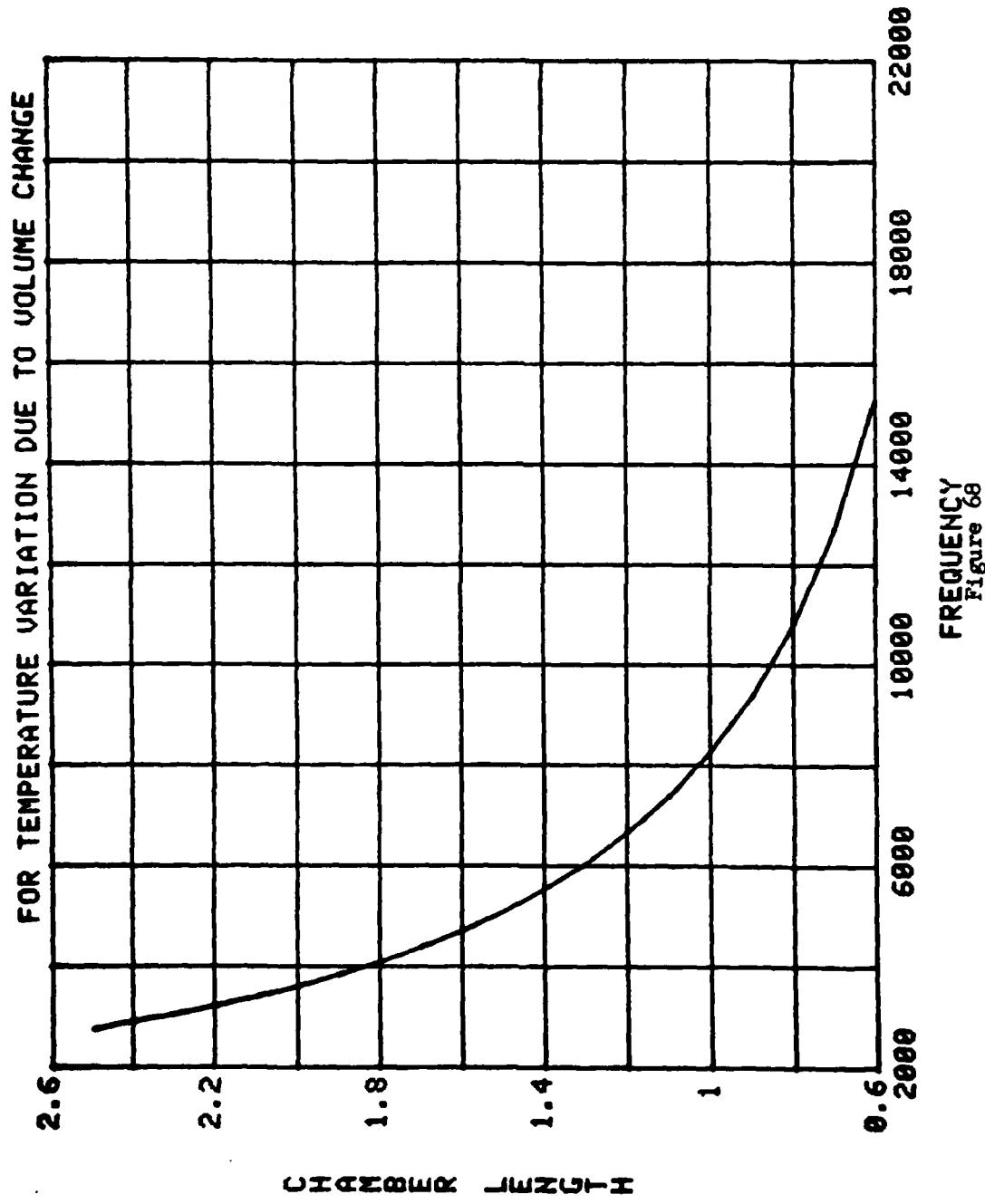


Figure 66

FREDDATA 30 Mar 81 14:32

| | |
|---------|-----|
| 15292.9 | 9.6 |
| 12710.3 | 9.7 |
| 10828.4 | 9.9 |
| 9401.16 | 1.1 |
| 8284.62 | 1.1 |
| 7389.27 | 1.1 |
| 6656.64 | 1.1 |
| 6047.01 | 1.1 |
| 5532.47 | 1.1 |
| 5992.88 | 1.1 |
| 4713.34 | 1.1 |
| 4382.62 | 1.1 |
| 4992.09 | 1.1 |
| 3935.03 | 1.1 |
| 3606.09 | 2.1 |
| 3401.02 | 2.1 |
| 3216.36 | 2.2 |
| 3049.29 | 2.3 |
| 2897.47 | 2.4 |
| 2758.95 | 2.5 |
| Ready | |

Figure 67



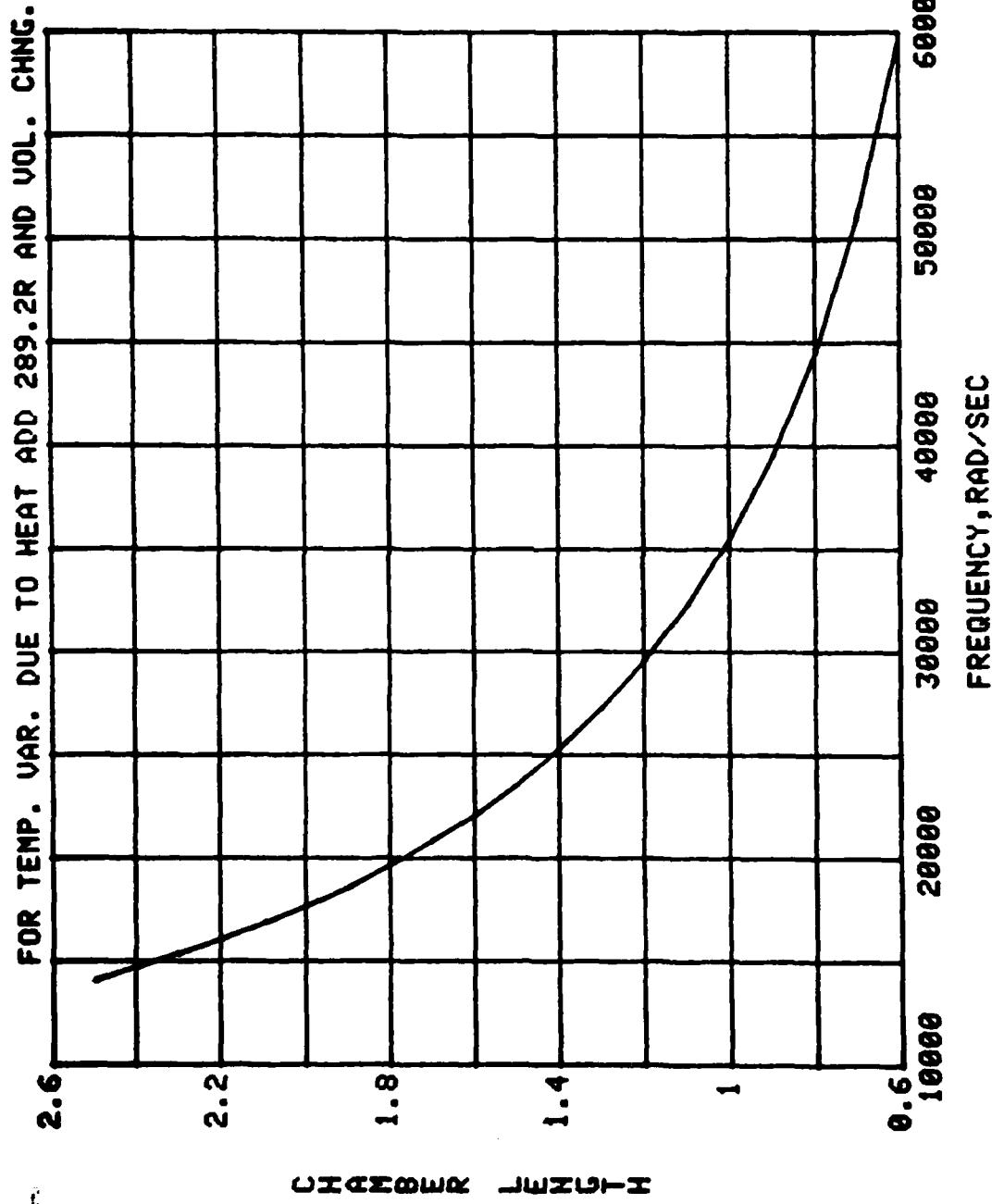


Figure 69

Appendix A

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SOLVE 04/01/81 10:26:11 PAGE: 1

```
100 REM THIS PROGRAM IN CONJUNCTION WITH ITS SUBROUTINES WILL SOLVE A
110 REM SECOND ORDER DIFFERENTIAL EQUATION OF THE FORM M*X''+R*X'+X/C=
120 REM P*COS WT, AND OUTPUTS A GRAPH OF DISPLACEMENT OR VELOCITY WITH
130 REM RESPECT TO TIME.
140 LIBRARY "L, IGSS: TEKSUB", "ERNOT", "VALUES"
150 FILE #1: "DATA1"
160 FILE #2: "DATA2"
170 SCRATCH #1
180 SCRATCH #2
190 PRINT "M*X''+R*X'+X/C= P*COS WT"
200 PRINT "DO YOU WANT TO INPUT EQUATION COEFFICIENTS OR PISTON DIMENSIONS (C OR D)?"
210 INPUT C?
220 IF C? = "C" THEN 250
230 CALL "VALUES", M, R, C, P, W
240 GO TO 360
250 PRINT "FOR THE ABOVE EQUATION INPUT M"
260 INPUT M
270 PRINT "INPUT R"
280 INPUT R
290 PRINT "INPUT 1/C"
300 INPUT C?
310 LET C = 1/C?
320 PRINT "INPUT P"
330 INPUT P
340 PRINT "INPUT W"
350 INPUT W
360 PRINT "HOW MANY CYCLES OF OSCILLATION DO YOU WANT SHOWN IN GRAPH?"
370 INPUT N
380 LET H = (PI*(W*100))/N
390 PRINT "DO YOU WANT A GRAPH OF DISPLACEMENT OR VELOCITY?"
400 PRINT "INPUT D OR V"
410 INPUT A?
420 LET H = A/(N*100)
430 REM THESE ARE THE INITIAL CONDITIONS
440 LET Y=Y1=Y2=S=V=0
450 PRINT S1: Y, Y2
460 PRINT S2: Y, Y2
470 LET S=1
480 LET Y=Y1
490 LET Y=Y2
500 FOR N=1 TO 4 STEPS
510 CALL "LNS01", M, R, C, Y1, Y2, W, P, T, F1, F2
520 LET K(N) = H#1
530 LET L(N) = H#2
540 IF N=74 THEN 620
550 LET Y1 = Y + K(N)/2
560 LET Y2 = Y + L(N)/2
570 LET T = S + H/2
580 IF N < 3 THEN 620
590 LET Y1 = Y + K(N)
600 LET Y2 = Y + L(N)
```

SOLVE

04/01/81

10:26:52

PAGE 2

```
610 LET T = S + H
620 NEXT N
630 LET Y1 = Y + ((K(1)+2*K(2)+2*K(3)+K(4))/6)
640 LET Y2 = Z + ((L(1)+2*L(2)+2*L(3)+L(4))/6)
650 IF T < R THEN 450
660 PRINT#1;T,Y1
670 PRINT#2;T,Y2
680 IF A$ = "D" THEN 710
690 IF A$ = "R" THEN 740
700 GO TO 390
710 RESET#1
720 CALL "TEKSUES":#1
730 GO TO 760
740 RESET#2
750 CALL "TEKSURE":#2
760 END
```

Appendix B

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EQMOT

04/01/81

10:44:41

PAGE 1

```
100 SUB"EQMOT":M,R,C,Y1,Y2,W,P,T,F1,F2
110 LET F1 = Y2
120 LET F2 = ((-R/M)*Y2-(1/(C*M))*Y1+(P/M)*COS(W*T))
130 SUBEND
```

Appendix C

10.28.55

04/01/81

VALUES

04/01/81

10:29:12

PAGE 1

```
100 SUB"VALUES":M,R,C,P,W
110 PRINT "INPUT THE CAP THICKNESS (IN INCHES)"
120 INPUT L
130 PRINT "INPUT THE CROSSECTIONAL AREA OF THE GAP (SQUARE INCHES)"
140 INPUT S
150 PRINT "INPUT THE CHAMBER VOLUME (CUBIC INCHES)"
160 INPUT V
170 PRINT "INPUT THE COMPRESSION RATIO"
180 INPUT G
190 PRINT "INPUT THE CAP WIDTH (INCHES)"
200 INPUT A1
210 LET A = A1/2
220 LET L1 = L + 1.2*A
230 LET T2 = (536*(G)^.4)
240 LET Q2 = (G^1.4)*14.7
250 LET J = Q2/(T2*53.3*12)
260 LET U = ((1.4*Q2.2*53.3*T2)^.5)*12
270 PRINT "DO YOU WANT TO INPUT W ? IF NOT THE RESONANT FREQUENCY WILL E"
280 PRINT "Y OR N"
290 INPUT B$
300 IF B$ = "Y" THEN 330
310 LET W = U*((S/(L1*V))^ .5)
320 GO TO 350
330 PRINT "INPUT W"
340 INPUT W
350 LET M = (J*L1)/S
360 LET K = W/U
370 LET R = ((J*U*K*K)/(2*3.14159))
380 LET C = V/(J*U*U)
390 LET Q = 2*3.14159*((L1*L1*L1*V/(S*S*S))^ .5)
400 PRINT "INPUT P"
410 INPUT P
420 PRINT "M =",M
430 PRINT "R =",R
440 PRINT "1/C =",1/C
450 PRINT "P =",P
460 PRINT "W =",W;"RAD/SEC"
470 PRINT "S =",S;"SQUARE INCHES"
480 PRINT "L =",L;"INCHES"
490 PRINT "V =",V;"CUBIC INCHES"
500 PRINT "2A =",2*A;"INCHES"
510 PRINT "Q =",Q
520 PRINT "COMPRESSION RATIO =",G
530 SUCEND
```

Appendix D

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| | | | | | | | | |
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| 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 0 | 0 | 0000 | 0 | 000 | 0 | 000 | 000 | 000 |

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|------|-------|-------|---|-----|---|----|------|
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| 0 | 0 | 0 | 0 | 0 | 0 | 00 | 00 |
| 0 | 0 | 0000 | 0 | 0 | 0 | 0 | 0 |
| 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 0 | 0 | 00000 | 0 | 0 | 0 | 0 | 0 |

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04/01/81

REVAMP

04/01/81

10:32:17

PAGE 1

```
100 REM THIS PROGRAM WILL ALLOW YOU TO STORE THE W VALUES AND THE
110 REM CORRESPONDING MAXIMUM DISPLACEMENT (3RD CYCLE) INTO A
120 REM DATA FILE, "DATA3". THIS THEN COULD BE USED WITH
130 REM OLD L. IG***:TEKGRAF3 TO PLOT THE FREQUENCY RESPONSE FOR
140 REM THE CERTAIN GEOMETRY
150 FILE #1;"DATA3"
160 PRINT "INPUT W"
170 INPUT W
180 PRINT "INPUT MAXIMUM DISPLACEMENT (3RD CYCLE)"
190 INPUT D
200 PRINT #1;W;";D
210 END
```

Appendix E

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| 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 00000 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
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| 00000 | 0000 | 00000 | 000 |
| 0 | 0 | 0 | 0 |
| 0000 | 0 | 0000 | 0 |
| 0 | 0000 | 0 | 0 |
| 0 | 0 | 0 | 0 |
| 0 | 0 | 0 | 00000 |

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FREQ

04/01/81

10:36:53

PAGE 1

```
100 REM THIS PROGRAM IS DESIGNED TO FIND THE FREQUENCY THAT A MACH WAVE
110 REM WILL OBTAIN IF PUT IN A COMBUSTION CHAMBER OF A CERTAIN VOLUME
120 REM COMPARED TO THE VOLUME AT THE TIME OF COMBUSTION.
130 REM IT WILL BE NECESSARY TO INPUT THE LENGTH OF THE COMBUSTION
140 REM CHAMBER AT THE TIME THE FREQUENCY IS SOUGHT FOR. THE INITIAL
150 REM CONDITIONS FOR THE CHAMBER AT THE BEGINNING OF COMPRESSION ARE
160 REM T0=536R, P0=14.7PSI. THE APPROXIMATION THAT THE WAVE IS
170 REM TRAVELING AT 1.1 TIMES THE SPEED OF SOUND THAT OCCURS IN THE
180 REM CHAMBER IS EVIDENT. OTHER ASSUMPTIONS ARE: ALL GASES IN THE
190 REM COMBUSTION CHAMBER ARE OF THE SAME CHARACTERISTICS AS AIR.
200 FILE#1:"FREQDATA"
210 SCRATCH#1
220 PRINT "INPUT ALL DISTANCES IN INCHES"
230 PRINT "INPUT THE COMPRESSION RATIO"
240 INPUT V
250 PRINT "INPUT THE BOTTOM DEAD CENTER CHAMBER LENGTH"
260 INPUT C
270 LET T2=(536*V)^.4
280 PRINT "INPUT THE TOP DEAD CENTER CHAMBER LENGTH"
290 INPUT L1
300 FOR L=L1 TO C STEP .1
310 LET A=49.01*12*((T2)^.5)
320 LET U=1.1*A
330 LET R=L/U
340 LET W=3.1415/R
350 PRINT "W IS"; W; "RAD/SEC"
360 PRINT#1:W; ", "; L
370 NEXT L
380 END
```

Appendix F

A repeating pattern of small 'X' marks on a white background.

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04/01/91

FREQ

04/01/81

10:28:18

PAGE 1

```
100 REM THIS PROGRAM IS DESIGNED TO FIND THE FREQUENCY THAT A MACH WAVE
110 REM WILL OBTAIN IF PUT IN A COMBUSTION CHAMBER OF A CERTAIN VOLUME
120 REM COMPARED TO THE VOLUME AT THE TIME OF COMBUSTION.
130 REM IT WILL BE NECESSARY TO INPUT THE LENGTH OF THE COMBUSTION
140 REM CHAMBER AT THE TIME THE FREQUENCY IS SOUGHT FOR. THE INITIAL
150 REM CONDITIONS FOR THE CHAMBER AT THE BEGINNING OF COMPRESSION ARE
160 REM T0=536R, P0=14.7PSI. THE APPROXIMATION THAT THE WAVE IS
170 REM TRAVELING AT 1.1 TIMES THE SPEED OF SOUND THAT OCCURS IN THE
180 REM CHAMBER IS EVIDENT. OTHER ASSUMPTIONS ARE: ALL GASES IN THE
190 REM COMBUSTION CHAMBER ARE OF THE SAME CHARACTERISTICS AS AIR.
200 FILE#1;"FREQDATA"
210 SCRATCH#1
220 PRINT"INPUT ALL DISTANCES IN INCHES"
230 PRINT"INPUT THE COMPRESSION RATIO"
240 INPUT V
250 PRINT "INPUT THE BOTTOM DEAD CENTER CHAMBER LENGTH"
260 INPUT C
270 LET T2=(536*V)^.4
280 PRINT "INPUT THE TOP DEAD CENTER CHAMBER LENGTH"
290 INPUT L1
300 FOR L=L1 TO C STEP .1
310 LET T3=(T2*(L1/L)^.4)
320 LET A=49.01*12*((T3)^.5)
330 LET U=1.1*A
340 LET R=L/U
350 LET W=3.1415/R
360 PRINT "W IS";W; "RAD/SEC"
370 PRINT#1;W;";L
380 NEXT L
390 END
```

Appendix G

AD-A109 656 NAVAL ACADEMY ANNAPOLIS MD F/G 20/13
OPTIMIZING THE NAHBE PISTON CAP DESIGN UTILIZING SCHLIEREN PHOT--ETC(U)
JUN 81 W H JOHNSON

UNCLASSIFIED USNA-TSPR-112

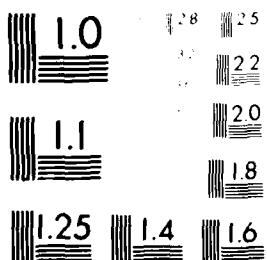
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04/01/81

EQ

04/01/81

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PAGE 1

```
10 REM THIS PROGRAM IS DESIGNED TO FIND THE FREQUENCY THAT A MACH WAVE
10 REM WILL OBTAIN IF PUT IN A COMBUSTION CHAMBER OF A CERTAIN VOLUME
10 REM COMPARED TO THE VOLUME AT THE TIME OF COMBUSTION.
10 REM IT WILL BE NECESSARY TO INPUT THE LENGTH OF THE COMBUSTION
10 REM CHAMBER AT THE TIME THE FREQUENCY IS SOUGHT FOR. THE INITIAL
50 REM CONDITIONS FOR THE CHAMBER AT THE BEGINNING OF COMPRESSION ARE
50 REM T0=536R, P0=14.7PSI. THE APPROXIMATION THAT THE WAVE IS
70 REM TRAVELING AT 1.1 TIMES THE SPEED OF SOUND THAT OCCURS IN THE
30 REM CHAMBER IS EVIDENT. OTHER ASSUMPTIONS ARE: ALL GASES IN THE
70 REM COMBUSTION CHAMBER ARE OF THE SAME CHARACTERISTICS AS AIR.
20 FILE#1: "FREQDATA"
10 SCRATCH#1
20 PRINT "INPUT ALL DISTANCES IN INCHES"
30 PRINT "INPUT THE COMPRESSION RATIO"
40 INPUT V
50 PRINT "INPUT THE BOTTOM DEAD CENTER CHAMBER LENGTH"
60 INPUT C
70 LET T2=(536*V)^.4
80 PRINT "INPUT THE TOP DEAD CENTER CHAMBER LENGTH"
90 INPUT L1
00 FOR L=L1 TO C STEP .1
10 LET T3=(T2*(L1/L))^ .4
20 LET T4=T3+289.2
30 LET A=49.01*12*((T4)^.5)
40 LET U=1.1*A
50 LET R=L/U
60 LET W=3.1415/R
70 PRINT "W IS";W; "RAD/SEC"
80 PRINT#1;W; ", ";L
90 NEXT L
00 END
```

